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EVALUATION OF A MECHANICAL SERVO-ACTUATOR FOR FLIGHT CONTROL SYSTEMS

LR 18636

By

O. A. Knuusi
W. R. Sage

October 1965

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-232(T)
LOCKHEED-CALIFORNIA COMPANY

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U. S. ARMY AVIATION MATERIEL LABORATORIES
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This report was prepared by the Lockheed-California Company in accordance with the requirements of Contract DA 44-177-AMC-232(T), initiated by the U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia.

This report represents the evaluation of the performance of a mechanical servo-actuator under loading conditions simulating the cyclic and collective pitch control functions of the Lockheed XH-51A rigid rotor helicopter.

This Command concurs with the contractor's conclusions that the currently designed mechanical servo-actuator is inferior to the hydraulic servo in performance, primarily because of excessive hysteresis and high force threshold. High noise level during operation was noted. The use of the currently designed servo-actuator for helicopter flight control systems is not practical until design improvements can be made and demonstrated.

This Command concurs in the recommendations that the research program for the application of this type of mechanical servo design principle for flight controls be continued.

Task 1P121401A141

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Prepared by
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Burbank, California

for

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ABSTRACT

A prototype mechanical servo-actuator was tested under simulated loads to determine its suitability for use in powering flight control systems for helicopters. Tests included frequency response, step response, threshold, hysteresis, endurance, and emergency operation, and were compared with similar tests of a typical hydraulic servo. Recommendations are made regarding design changes for further investigation.

FOREWORD

This report pertains to work performed under U. S. Army Aviation Materiel Laboratories (USAAVLABS)* Contract DA 44-177-AMC-232(T) dated 30 June 1964, by the Lockheed-California Company as prime contractor and the Curtiss Division of Curtiss-Wright Corporation as subcontractor. Mr. M. B. Salomonsky of USAAVLABS served as the authorized representative of the Contracting Officer for this program.

The test article was designed and built by Curtiss-Wright under the direction of John S. Perryman, Chief Project Engineer of the Curtiss Division. The Lockheed effort was directed by O.A. Knuusi, R & D Engineer, Flight Control Systems Department; the testing was done at the Lockheed Rye Canyon Research Center by W. R. Sage, Senior Research Engineer, Vehicle Systems Laboratory, with the assistance of R. E. Colvin, Curtiss-Wright Design Engineer. The report was prepared by O. A. Knuusi and W. R. Sage of the Lockheed-California Company as co-authors.

*Formerly, U. S. Army Transportation Research Command

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SUMMARY

This report discusses the results of a test program performed by the Lockheed-California Company under an USAAVLABS contract to evaluate the performance of a mechanical servo-actuator under loads simulating a helicopter flight control system. The mechanical servo tested was designed and fabricated by Curtiss-Wright to a design specification prepared under this contract.

The test item was subjected to a series of tests including force threshold, resolution, frequency response, step response, and endurance, and the results were compared to similar tests of a typical hydraulic servo.

The performance of the mechanical servo indicated that two characteristics needed correction: excessive dead-band and high force threshold. After completion of the scheduled test program, the mechanical servo was returned to Curtiss-Wright for investigation of possible design changes.

The mechanical servo was modified by Curtiss-Wright and returned to Lockheed-California. One day of testing indicated that the changes made by Curtiss-Wright had reduced the dead-band to 10 percent of its former value and had reduced the force threshold at null by 25 percent. These performance changes were made by compromising the load-holding function of the spring clutch.

Curtiss-Wright also submitted proposal layouts of an improved design which would provide an irreversible screw jack for the braking function and simple linkages to replace the input and output gear trains of the previous design.

It is concluded that the mechanical servo as presently configured does not perform well enough under the test conditions to match the performance of the hydraulic servo, primarily because of excessive hysteresis and high-force threshold. High noise level during operation under heavy loads was also noted.

Retest of the modified unit and a study of proposed design changes indicate that significant improvement in performance can be achieved from a redesigned servo.

Further testing of an improved design under conditions more representative of actual service is considered necessary to more completely evaluate the potential use of mechanical servos in flight control systems.

CONCLUSIONS

The results of the test program indicate, in general, that the prototype mechanical servo will perform the tasks for which it was designed but not as well as the hydraulic servo to which it was compared. The principal areas in which the mechanical servo appears to be inferior to the hydraulic servo are:

1. Force Threshold. The force required to provide an initial output of the mechanical servo against typical loading was approximately 5 pounds as compared to 1/4 pound for the hydraulic actuator.

2. Hysteresis, or dead-band. The mechanical servo, as tested, indicated a total dead-band of approximately 2 degrees, or 2.7 percent of total amplitude, as compared to 0.3 percent for the hydraulic actuator.

3. Resolution. The capability of the mechanical servo to respond to small increments of input against a load proportional to displacement was inconsistent, especially under high loads.

4. Noise Level. The mechanical servo operation was very noisy against high loads, compared to the hydraulic servo. Admittedly, most of the noise was generated by the flexible shaft connection between the electric motor and the servo, but since the intended application of the servo is based on the use of a flexible-shaft power drive, the noise problem is a real one.

It is therefore concluded that the use of a mechanical servo-actuator of this type in helicopter flight control systems is not practical until design improvements can be made and demonstrated.

RECOMMENDATIONS

It is recommended that research programs for the application of mechanical servo design principles to flight control systems be continued, for the following reasons:

1. The basic principle of utilizing mechanical power directly without converting the energy to other media has obvious advantages (weight saving, space saving, high reliability) which should be considered in any trade-off study of powered control systems.

2. The reliability of mechanical components should be less affected by high-temperature environment than hydraulic or electrical components.

3. The performance discrepancies noted in this evaluation are of a nature which can be corrected by rational evaluation of the causes and by redesign of the components to minimize or eliminate the problem areas.

This investigation was limited to the evaluation of a single mechanical actuator operating against simulated loading but with no attempt to provide facilities for pilot evaluation of a multi-axis mechanical system. We therefore recommend that a follow-on program be considered to evaluate the performance of mechanical servo-actuators in a three-axis (pitch, roll, and lift) helicopter control system installed on a whirl tower which will provide more representative dynamic loading and will include provisions for pilot evaluation of the system. The proposed program will require three servo-actuators, one each for the pitch and roll cyclic control and one for collective pitch (lift) control.

It is recommended that the mechanical servo-actuators procured for such a program be designed to correct or minimize the deficiencies noted in the evaluation of the prototype actuator and that the reliability, serviceability, weight, and structural integrity of the actuators be adequate for installation in a flying vehicle.

INTRODUCTION

The use of mechanical servo-actuators in flight control systems has been proposed and investigated in earlier studies (References 1 and 2). The results have generally indicated that performance under specific load conditions needs further study but that mechanical systems can be expected to show significant savings in overall weight and have a potential advantage in achieving high reliability under adverse environmental conditions. The realization of high-quality performance of specific control functions requires intensive testing of hardware designed for the purpose.

This program was established to evaluate the performance of a mechanical servo-actuator under loading conditions simulating the cyclic and collective pitch control functions of the Lockheed XH-51A rigid rotor helicopter. The work statement specified the preparation of a design specification for the mechanical servo-actuator, the fabrication of a prototype actuator, the preparation of a test procedure, the fabrication of a test fixture, the performance testing of both the mechanical servo-actuator and an equivalent hydraulic servo-actuator, and the preparation of a test report.

BASIS FOR DESIGN OF TEST ITEM

In order to have an equitable basis for evaluation of a mechanical servo-actuator, the design specification (see Appendix) was based on the requirements of the cyclic and collective pitch control functions of the Lockheed XH-51A rigid rotor helicopter. This vehicle presently uses three hydraulic servos for these functions. The two cyclic control servo-actuators operate against high-rate springs which transmit forces proportional to actuator displacement to the swash plate and in turn to a control gyro, the precession of which modulates the blade pitch angle. The collective pitch actuator controls the vertical displacement of the gyro with reference to the rotor plane, which requires it to operate against the mass of the swash plate, gyro, and associated linkage.

The three hydraulic actuators all have the same net piston area, but the stroke of the collective actuator is 4.35 inches as compared to 2.03 inches stroke for the cyclic actuators. In order to have a single mechanical servo design perform both control functions, a compromise was necessary. The design specification shown in the Appendix required that the mechanical servo accept input displacements and provide output displacements for both functions, but the output force requirements were based on the cyclic control loading.

The space envelope and the orientation of input and output connections for the mechanical servo were dictated by the desire to build a unit suitable for possible future installation in the XH-51A vehicle control system. However, it was recognized that any future program for airplane installation would require additional units and probably redefinition of functional requirements, so that overall length, weight, and details of linkage connections were compromised in the interests of expediency.

The location of the power input connection was specified to accommodate a relatively straight flexible shaft for the transmission of power from the existing hydraulic pump pad on the XH-51A transmission at 6300 rpm.

The design specification is included in this report as the Appendix.

DESCRIPTION OF TEST ITEM

The mechanical servo-actuator was designed and built by the Curtiss Division of the Curtiss-Wright Corporation to the requirements of the design specification (see Appendix). The input and output lever travel is limited by fixed stops to total angular displacement of 75 degrees. The output lever, 90 degrees displaced from the input lever, moves in phase with the input lever except for the dead-band required to energize the spring clutch.

The operation of the mechanical servo can best be described by reference to Figure 1. The power input, at 6300 rpm, is transmitted through a worm gear and spur gearing to two cylindrical drums which rotate at 180 rpm. Spring clutches, geared to the output elements, have a small clearance to the drum surface and normally bear against an internal cylindrical steel surface to provide a brake to hold the output load. Angular motion of the input lever is transmitted and amplified by a gear train to the end of one of the spring clutches. When the spring clutch contacts the rotating drum, the self-energizing clutch transmits the available torque of the rotating drum to the output lever through reduction gearing. The clutch releases as soon as the output has moved to a position commanded by the input signal. When the clutch is energized, the output moves at a rate proportional to the power input rpm, regardless of the rate of the input arm. Consequently, an input command at a rate slower than maximum results in a "stair-step" output response, since each time the output catches up with an increment of input command, the clutch releases until re-energized by continuing motion of the input. The gear ratio between the input shaft and the spring clutch is 11.6; consequently the input or output rate corresponding to the 180-rpm drum speed is 15.5 rpm. At this rate, full travel (75 degrees) requires 0.8 second.

Both the input lever and the output lever have attachment holes to provide the required linear travel for the two loading conditions (cyclic and collective) within the 75-degree travel limit.

The mechanical servo includes provisions for manual override in the event of power failure or clutch release failure. The input and output shafts are concentric and normally move in phase. The input shaft motion required to energize the servo clutch is approximately one degree. The input torque is transmitted to the input gear train through a preloaded spring bungee. In the event of power failure, the spring bungee permits continued motion of the input lever, which is transmitted to the output lever by a pin which normally is in the center of a slotted hole. Motion of this pin relative to the output also releases one of a pair of clutch springs which normally transmits power to the output shaft. The input

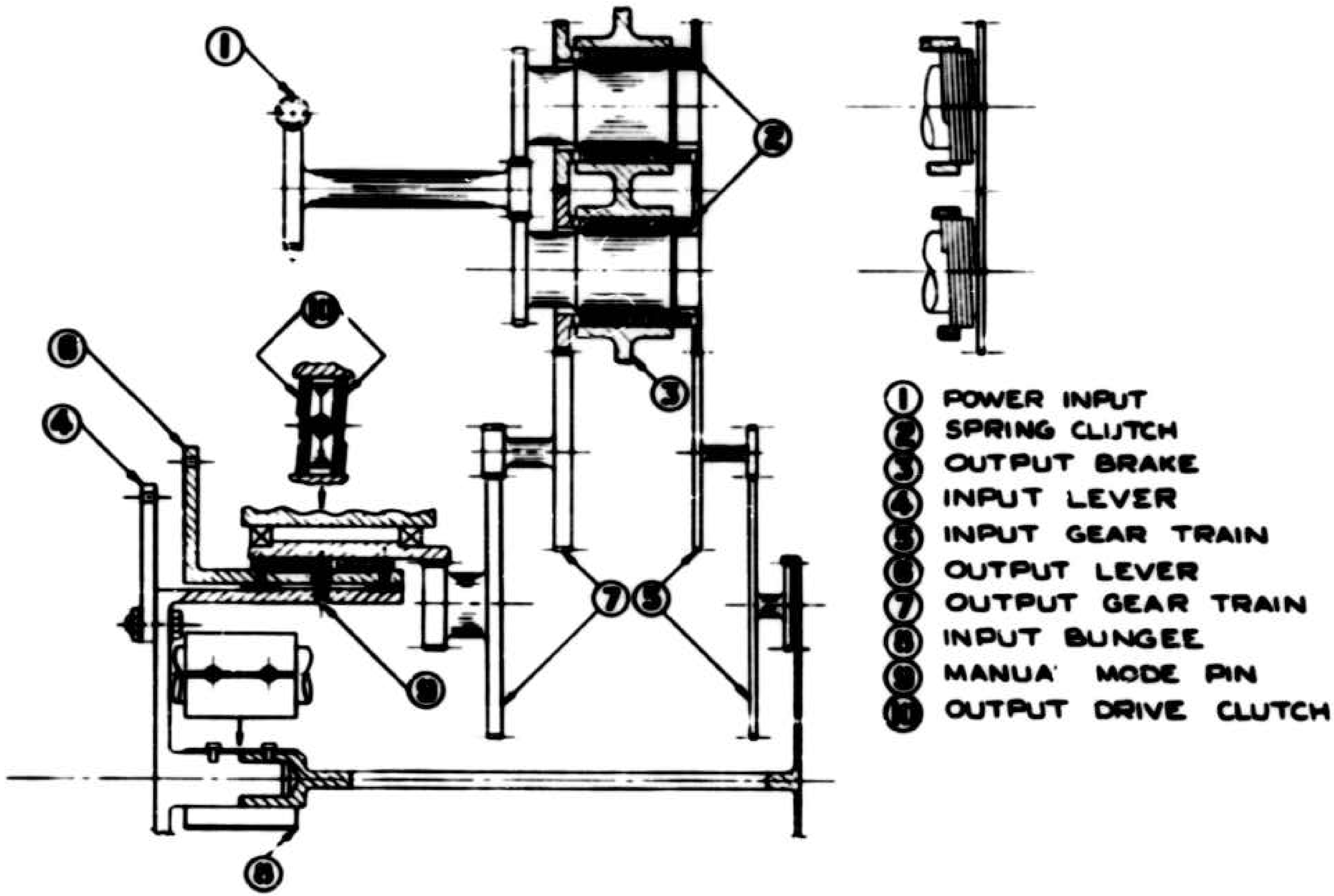

FIGURE 1 SCHEMATIC DIAGRAM - MECHANICAL SERVO

shaft thus drives the output shaft in direct ratio with the input bungee torque (20 to 30 inch-pounds) and the friction of the overrunning output clutch added to the load.

A servo failure causing the output to continue motion without an input command would result in a phase differential between input and output; consequently the aforementioned pin would release the output drive clutch and connect the output load directly to the input.

A more detailed description of the mechanical servo-actuator and its components appears in Reference 3.

DESCRIPTION OF TEST FIXTURE

A test fixture was designed and fabricated for the performance of the tests prescribed by Reference 4. The fixture included mounting provisions for both the mechanical servo and the hydraulic servo, an output loading bell crank for the application of both spring loads and inertia loads as specified by Reference 4, a hydraulic actuator controlled by an electro-hydraulic transfer valve to provide the required programmed inputs, and a d.c. electric motor as the power source. A 3-foot flexible shaft was provided to transmit power to the mechanical servo. A flywheel to simulate the inertia of the XH-51A rotor was provided but was not used, as it would require too much time to stop the flywheel in the event of malfunction of the mechanical servo-actuator.

Instrumentation included a linear potentiometer to read input amplitude, a force transducer to read input load, a rotary potentiometer to read output amplitude, a strain gauge on the load bell crank to read output load, and a 4-channel Sanborn recorder (Model No. 154-100 BP) to plot the values of the above four parameters as a function of time.

A function generator (Hewlett-Packard Model No. 202A) was provided to program the amplitude and frequency of the input signals.

Auxiliary instrumentation included a stroboscope to monitor power input rpm, a temperature indicator to monitor servo body temperature, spring scales and torque wrenches for direct force measurements, and 0.001-inch division dial gauges for precise reading of small amplitudes.

TEST PROCEDURE AND RESULTS

MECHANICAL SERVO-ACTUATOR

The mechanical servo-actuator, Curtiss-Wright Corporation Part No. 173410, Serial No. 1, was mounted in a loading test fixture as per Figure 2 and as shown in Figures 3 through 7.

Two types of loads were separately attached to the actuator - cyclic and collective. The cyclic loads consisted of three different spring loads being individually applied. These springs were alternately attached to the output arm of the mechanical servo-actuator at a radius of 2.178 inches. The springs had the following rates:

1. Heavy Spring - 510 pounds per inch.
2. Medium Spring - 155 pounds per inch.
3. Light Spring - 47 pounds per inch.

The collective loads consisted of four different loading configurations which were alternately attached to the output arm of the mechanical servo actuator at a radius of 3.622 inches. The loading configurations consisted of the following springs and masses:

1. 50-pound/inch spring and 20-pound mass.
2. 50-pound/inch spring and 40-pound mass.
3. 10-pound/inch spring and 20-pound mass.
4. 10-pound/inch spring and 40-pound mass.

Figure 8 shows a dimensional relationship of the load bell crank and the attached 20-pound and 40-pound masses.

The variation in loading was provided in order to determine whether the magnitude of the load had a significant effect on performance.

After the mechanical servo-actuator was mounted in the test fixture, the collective loading configuration of 10-pound/inch spring and 20-pound mass was attached to the actuator. During the preliminary manual operation of the actuator, the output loading arm encountered a mechanical

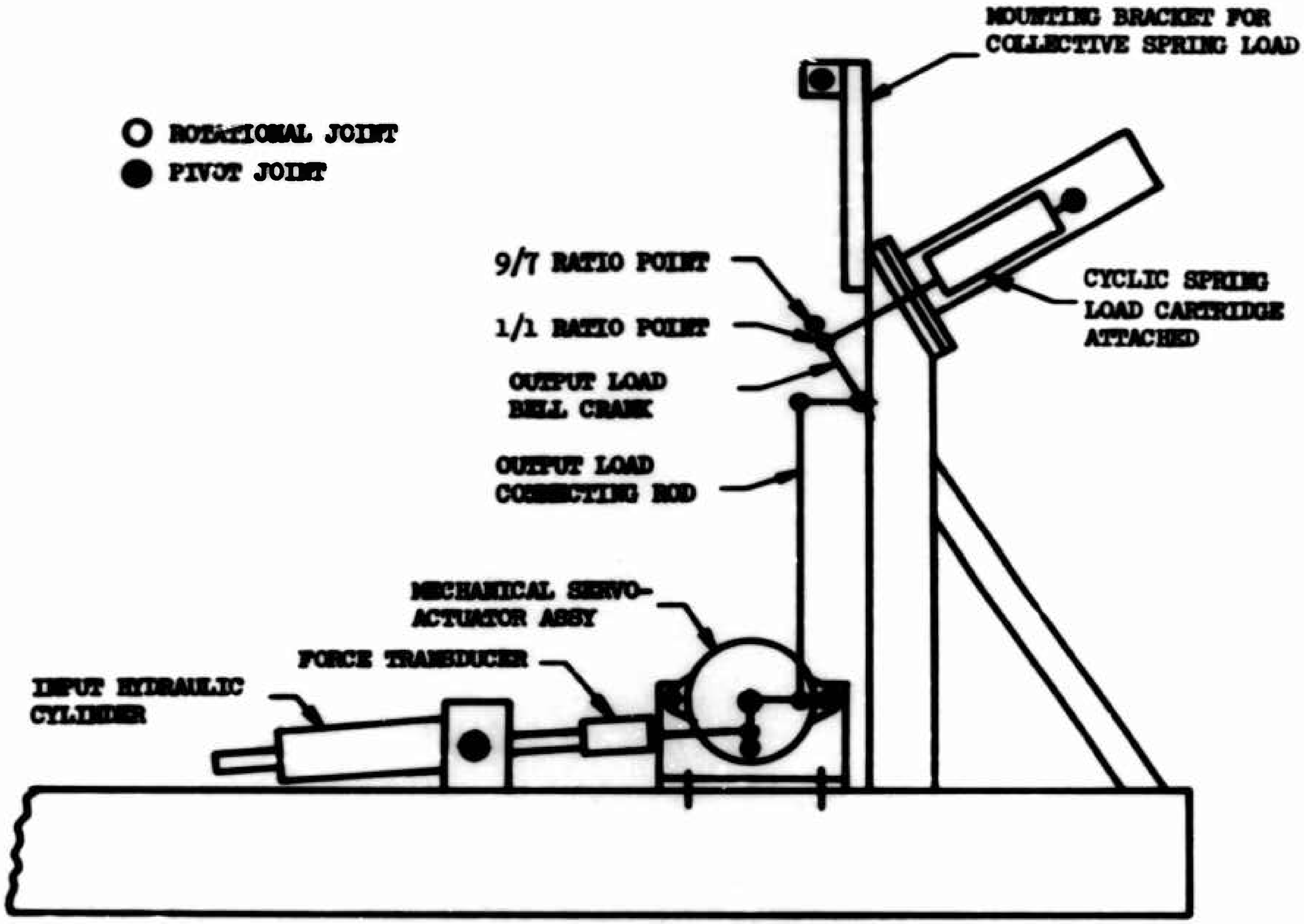

FIGURE 2 SCHEMATIC DIAGRAM OF MECHANICAL SERVO-ACTUATOR INSTALLED IN TEST FIXTURE

FIGURE 3 MECHANICAL SERVO, CYCLIC LOAD, FULL CLOCKWISE

FIGURE 4 MECHANICAL SERVO, COLLECTIVE LOAD, FULL CLOCKWISE

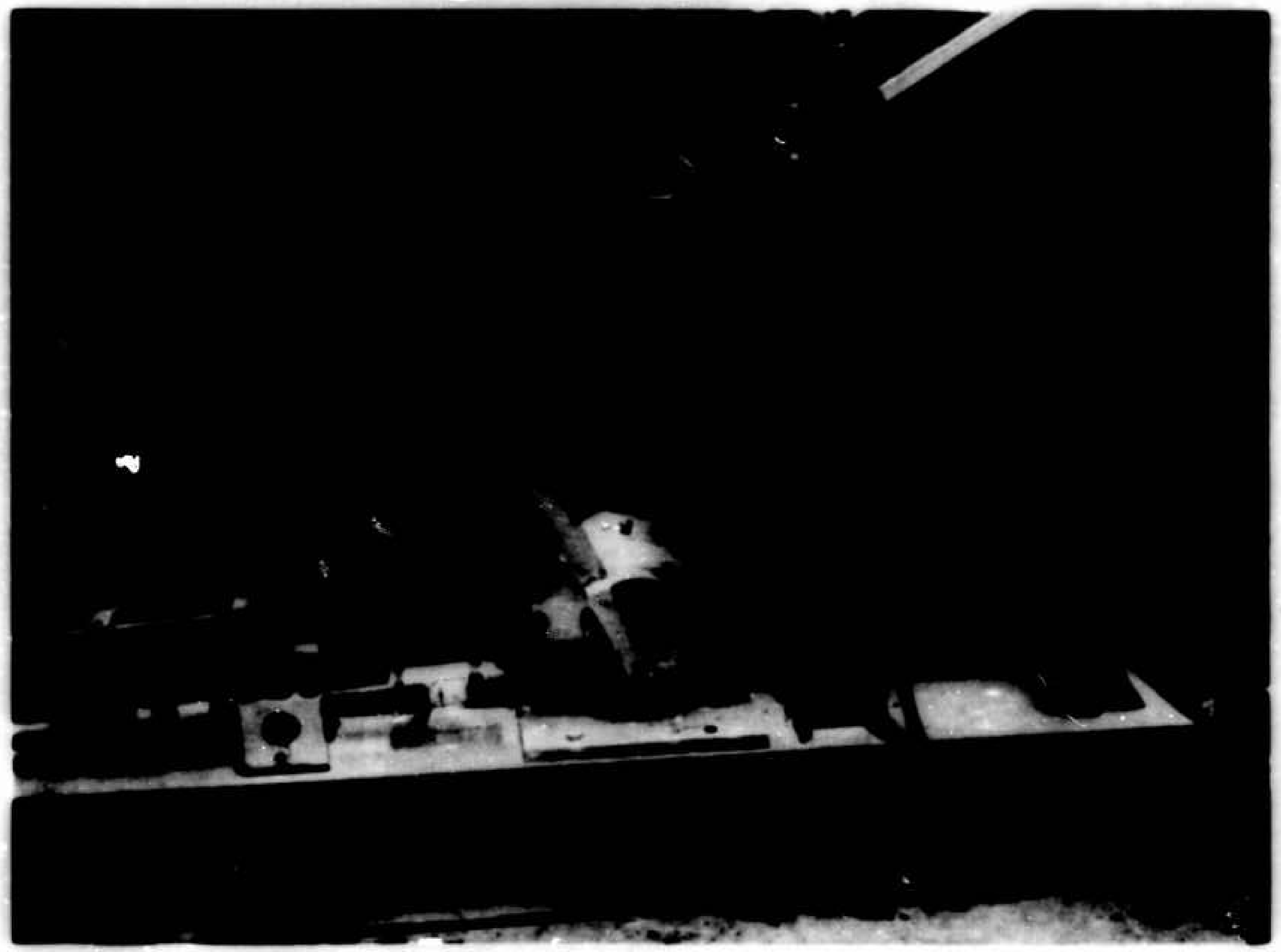

FIGURE 5 MECHANICAL SERVO, COLLECTIVE LOAD, MID-POSITION

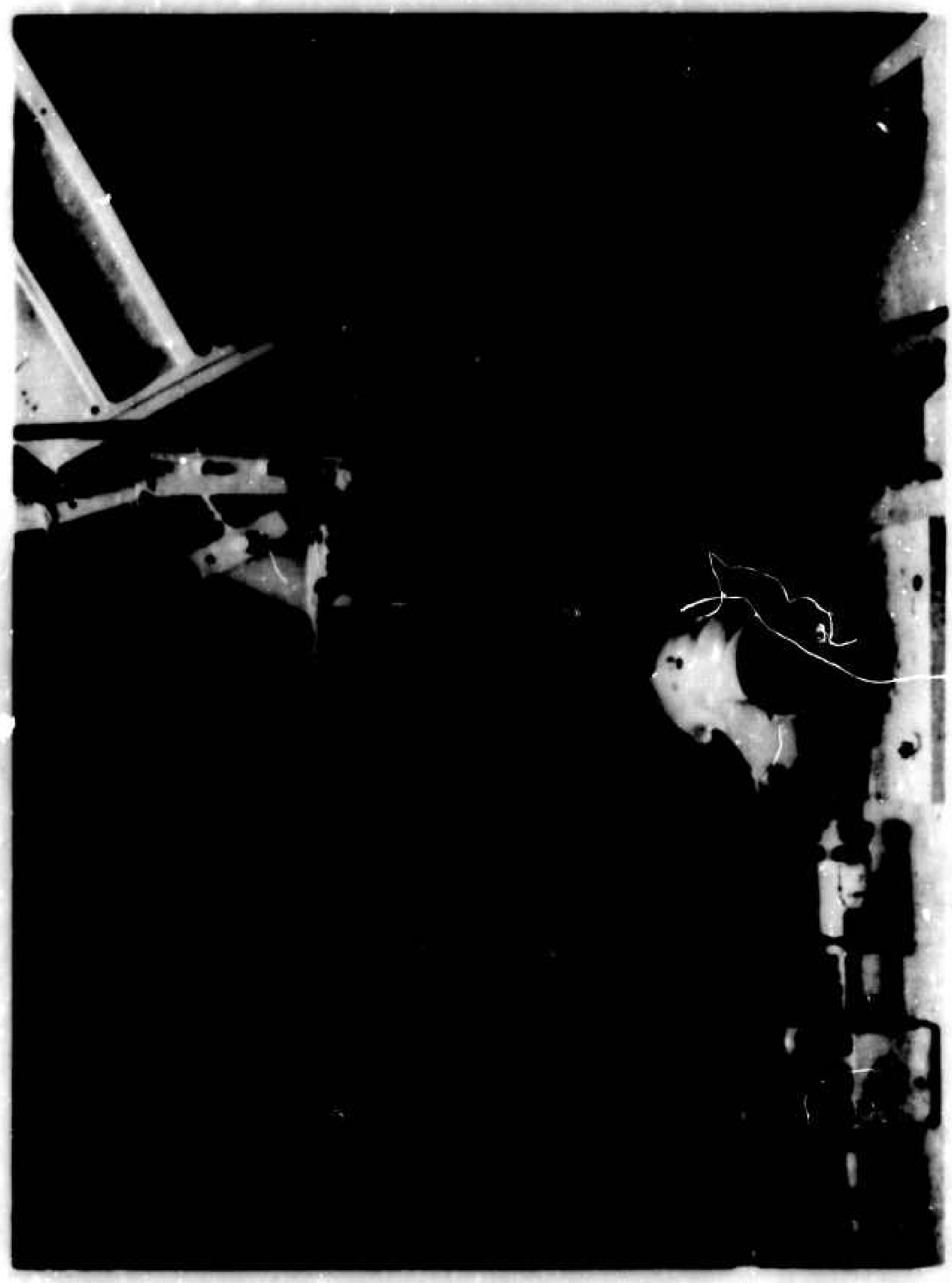

FIGURE 6 MECHANICAL SERVO, CYCLIC LOAD, NEUTRAL POSITION

FIGURE 7 MECHANICAL SERVO, CYCLIC LOAD, GENERAL VIEW

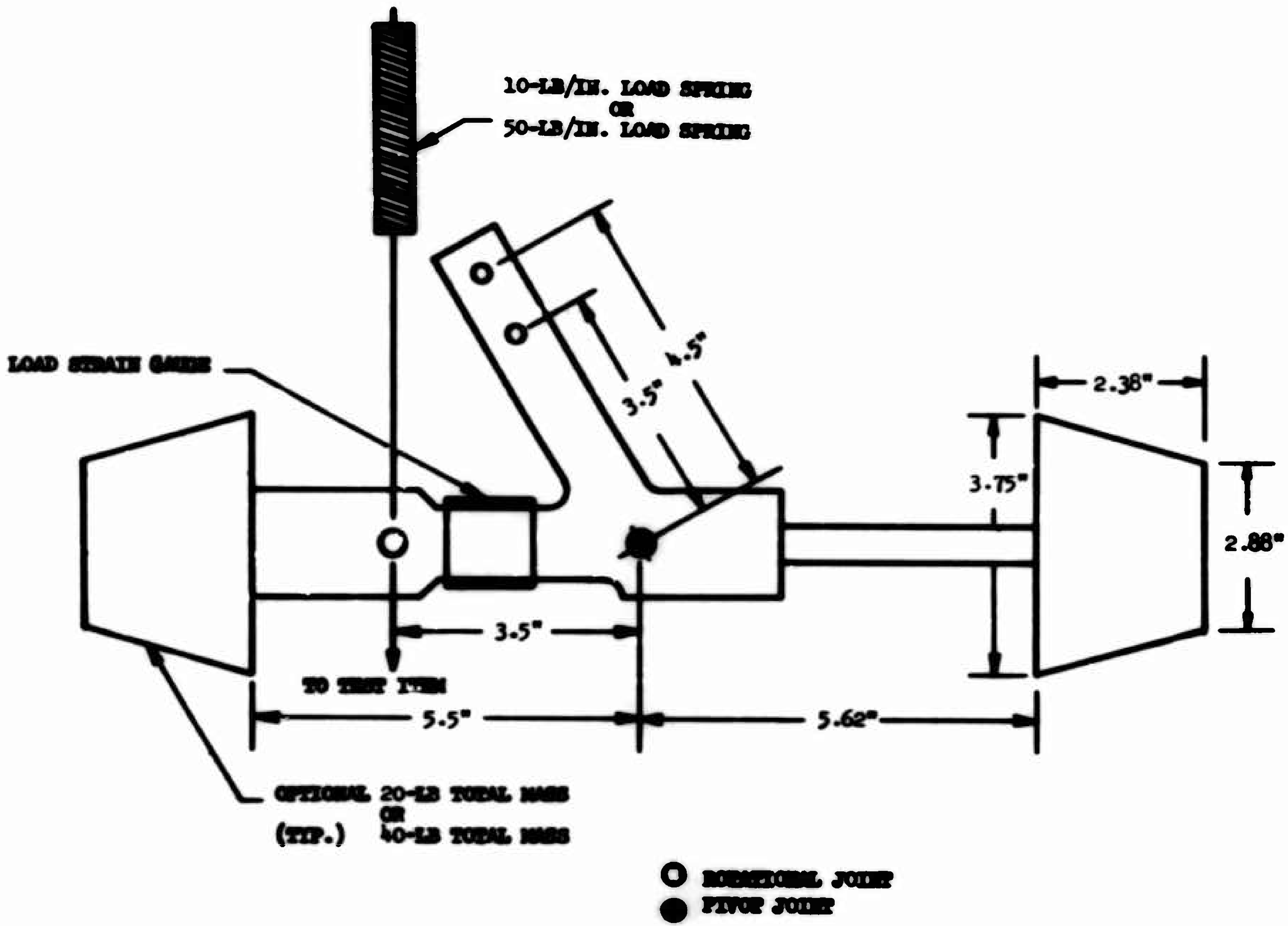

FIGURE 8 COLLECTIVE LOAD CONFIGURATION

stop on the test fixture which caused the 6300-rpm input power flexible cable to break. Since this was an initial checkout of the actuator, the input power source was running at only 3000 rpm when the flexible cable failed.

The mechanical servo-actuator was then operated in the "emergency" manual power mode. During the initial clockwise motion of the output arm, a failure occurred within the actuator. At the time of the failure, the 10-pound/inch collective load spring was extended approximately 2 inches and the input force load transducer was indicating a force of 100 to 120 pounds.

Disassembly of the mechanical servo-actuator by Curtiss-Wright Corporation personnel revealed that the bungee spring T-bracket had broken. The T-bracket was replaced and the actuator remounted in the test fixture. A relief valve was added to each of the input hydraulic cylinder lines to limit the input force to the mechanical servo-actuator to approximately 70 to 80 pounds. Also the input hydraulic cylinder was attached to the input arm 1.562-inch radius attachment point for all further cyclic and collective tests. After these changes were made, the following operational tests were performed with the input power applied at 6300 rpm.

Cyclic Load - Force Threshold Test

Force threshold is defined as the minimum input torque required to cause motion of the output arm of the mechanical servo-actuator. The applied input torque was measured by a standard laboratory-type torque wrench. The output motion of the actuator output arm was determined by placing a 0.001-inch division dial gauge against the output arm vertical connecting rod. With the 510-pound/inch load spring attached, the output arm was moved from neutral to 0.25-, 0.50- and 0.90-inch stroke positions, both clockwise (CW) and counterclockwise (CCW). At each of these stationary output stroke positions, the force threshold was measured in both clockwise and counterclockwise directions. This test was also repeated following the endurance cycling test. The results of the tests are tabulated in Table 1.

It was noted that the force threshold varied from 10 to 20 inch-pounds when the output arm stroke was being increased and from 9 to 18 inch-pounds when decreasing the output arm stroke. In general the output arm motion was smooth or in small step increments (0.003 to 0.070 inch) when the stroke was being increased; however, the output arm motion was in definite steps varying from small steps of 0.030 inch to large steps of 0.600 inch when decreasing the output stroke. The only appreciable change noted after the endurance cycling was that the output arm motion was noted to change to larger steps.

TABLE 1

FORCE THRESHOLD TEST RESULTS

Output Stroke Direction	Output Stroke Inches	Applied Force to Input Arm Direction	Applied Force to Input Arm Torque, in.-lb	Motion of Output Arm, in.
		510-Pound/Inch Cyclic Spring		
Neutral	0	CW	11	Smooth
		CCW	11	.030 steps
CW	.25	CW	18	Smooth
		CCW	9	.060 steps
CW	.50	CW	18	Smooth
		CCW	10	.060 steps
CW	.90	CW	20	.030 steps
		CCW	16	.10 steps
CCW	.25	CW	11	.20 steps
		CCW	11	.015 steps
CCW	.50	CW	12	.3 steps
		CCW	12	Smooth
CCW	.90	CW	14	.4 steps
		CCW	18	Smooth
		10-Pound/Inch Spring and 20-Pound Mass Collective Load		
CW	.01	CW	8	Smooth
		CCW	10	Smooth
CW	.4	CW	10	Smooth
		CCW	13	.003 steps
CW	1.0	CW	10	Smooth
		CCW	8	.030 steps
CW	2.0	CW	13	Smooth
		CCW	7	.040 steps
		After Endurance Test, 510-Pound/Inch Cyclic Spring		
Neutral	0	CW	10	Smooth
		CCW	14	Smooth
CW	.25	CW	12	.030 steps
		CCW	9	.060 steps
CW	.50	CW	10	.070 steps
		CCW	11	.3 steps
CW	.90	CW	17	.050 steps
		CCW	18	.6 steps
CCW	.25	CW	10	.25 steps
		CCW	16	.020 steps
CCW	.50	CW	11	.5 steps
		CCW	14	.025 steps
CCW	.90	CW	9	.6 steps
		CCW	16	.015 steps
		10-Pound/Inch Spring and 20-Pound Mass Collective Load		
CW	2.0	CW	9	Smooth
		CCW	7	.040 steps

Cyclic Load - Resolution Test

Resolution is defined as the minimum input motion required to cause motion of the output arm. The input motion was measured by placing a 0.001-inch division dial gauge against the force load transducer which was directly connected to the mechanical servo-actuator input arm 1.562-inch-radius attachment point. The output motion of the actuator output arm was measured by placing a 0.001-inch division dial gauge against the output arm vertical connecting rod. The input hydraulic cylinder rod was moved by means of a manually rotated screw drive.

Prior to starting these measurements, the mechanical servo-actuator was noted to have a dead-band stroke of ±0.050 inch as measured at the 2.326-inch-radius attachment point of the input arm. This is equivalent to approximately 2.5-degree total dead-band rotation of the input shaft.

To eliminate the effect of the dead-band, the input arm was slowly rotated in one direction until the output arm moved. At this point, the input arm motion to cause a second output arm motion was measured and recorded. Measurements were made with each of the attached load springs at neutral and at strokes of 0.25, 0.50 and 1.00 inch,both clockwise and counter-clockwise. It was noted that as the output load arm stroke was increased, the resulting output arm motion was created as varying step changes instead of as a smooth motion. The cyclic load spring (510-pound/inch) test was repeated after the endurance cycling test. Results of these tests are tabulated in Table 2. No appreciable change was noted after the endurance test.

Cyclic Load - Maximum Rate Test

The mechanical servo-actuator was operated by applying a square-wave electrical signal to the servo loop of the input hydraulic cylinder. Only the light spring (47-pound/inch) cyclic load condition rate test appeared valid as the relief valves were opening when the medium (155-pound/inch) and the heavy (510-pound/inch) spring rate tests were performed.

After the completion of the endurance cycling test, the maximum rate test was rerun using the heavy (510-pound/inch) spring. The input was manually rotated via a 10-inch lever arm attached to the input shaft. Results of these tests are tabulated in Table 3.

Cyclic Load - Frequency Response Test

The mechanical servo-actuator was cycled sinusoidally at output amplitudes of ±0.05, ±0.10 and ±0.50 inch. The frequency was varied from 0.1 cps to a maximum of 10 cps, depending on the capability of the actuator. The basic limitation to obtaining the higher frequencies was the previously mentioned hydraulic relief valves. Frequency response measurements were made using each of the three load springs. After the endurance cycling test, the frequency response test was performed using the heavy (510-pound/inch) spring at an amplitude of ±0.10 inch. This test was repeated with the hydraulic relief valves installed (R.V. IN) and without hydraulic relief valves installed (NO R.V.) at frequencies ranging from 0.1 cps to 8 cps. The results of these tests are presented in Figures 9 through 19.

During these tests, the motions of the signal input arm and the output load bell crank were recorded on a Sanborn recorder. The magnitude of each of the corresponding amplitude traces was measured, and the Amplitude Ratio, expressed in decibels (DB), was determined by using the following equation:

$$\text{Amplitude Ratio} = 20 \log_{10} \frac{\text{Output Amplitude}}{\text{Input Amplitude}}$$

Figures 9 through 19 also show the phase lag in degrees between input and output.

Cyclic Load - Step Response Test

The mechanical servo-actuator was operated at maximum rate with output steps of 0.10, 0.25 and 0.50 inch for each of the three cyclic load springs. Following the endurance cycling test, the heavy (510-pound/inch) spring was attached to the output arm, and the actuator was subjected to a step response test. Incremental steps of 0.10 inch and 0.25 inch were applied without the hydraulic relief valves installed. The time constant was determined for each step condition,and the results are tatulated in Table 4. The time constant is defined as the time in seconds for the output arm to travel 63 percent of its full step travel. Also, since the input step motion of the input hydraulic cylinder is not a true square-wave step, the tabulated time constant is the difference in time for the input hydraulic cylinder and the output arm of the actuator to reach their respective 63-percent points of total travel.

Collective Load - Force Threshold Tests

The 10-pound/inch load spring and the 20-pound mass were attached to the output arm. The force threshold measurements were made at strokes of 0.01, 1.00, 2.00 and 3.60 inches. After completion of the endurance cycling tests, the force threshold measurement was made only at the 2.00-inch stroke. Results of these tests are tabulated in Table 1.

Collective Load - Resolution Tests

Resolution was measured using the same technique as described in the cyclic load section. Measurements were made with the 10-pound/inch spring and 20-pound mass and with the 50-pound/inch spring and 20-pound mass attached to the output arm. Strokes of 0.01, 1.00, 2.00 and 3.60 inches were used. After completion of the endurance cycling tests,a resolution measurement was made only at the 2.00-inch stroke position with the 10-pound/inch spring and 20-pound mass attached. Results of these tests are tabulated in Table 2.

Collective Load - Maximum Rate Tests

Maximum rate tests were performed as described in the cyclic load section for each of the four loading conditions. After the completion of the endurance cycling test, a maximum rate test was rerun with 10-pound/inch spring and 20-pound mass attached to the output arm of the mechanical servo-actuator. The input arm of the actuator was manually moved for this one test. Results of these tests are tabulated in Table 3. It was noted

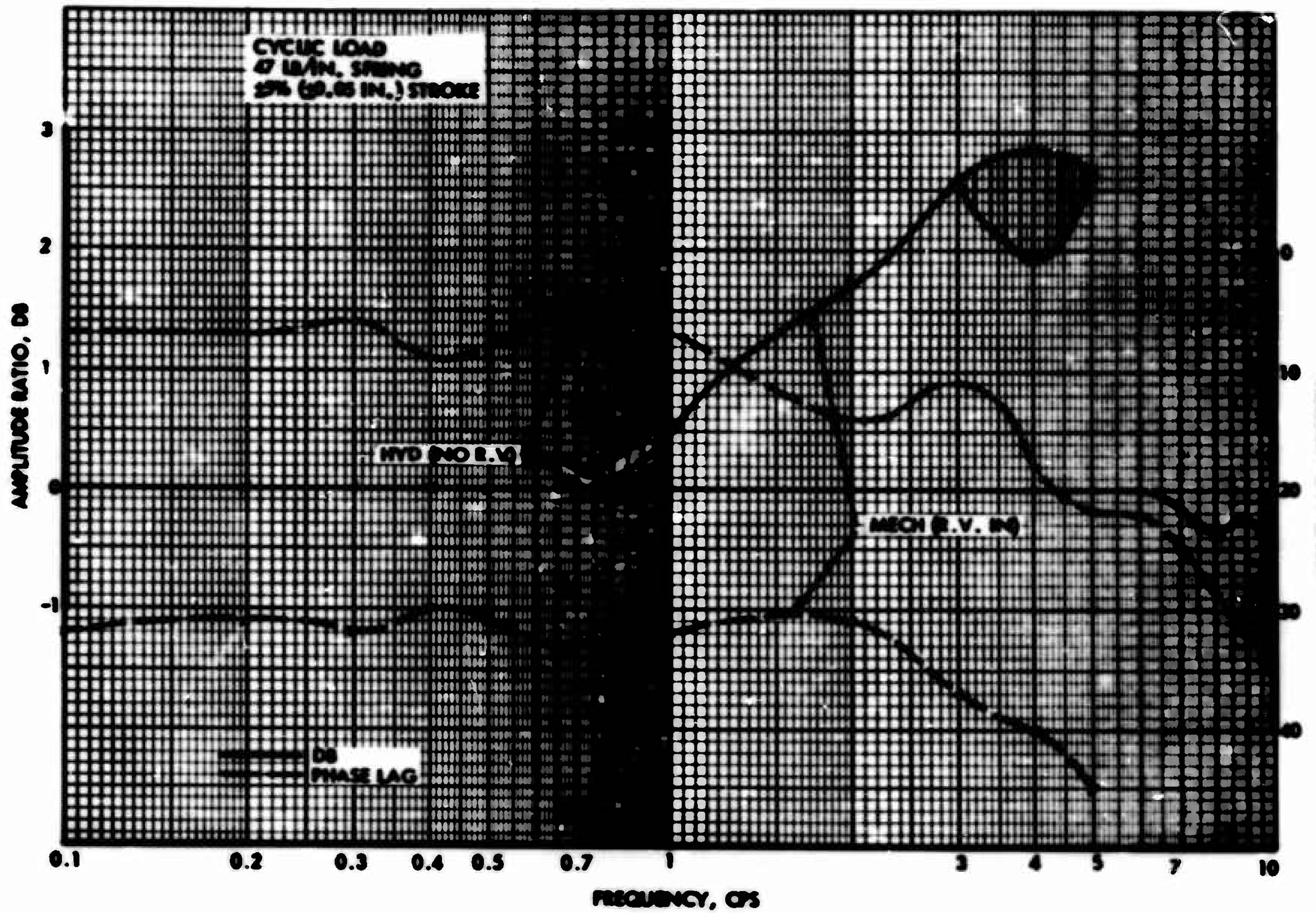

FIGURE 9 FREQUENCY RESPONSE, CYCLIC LOAD, 47-POUND SPRING, 5-PERCENT STROKE

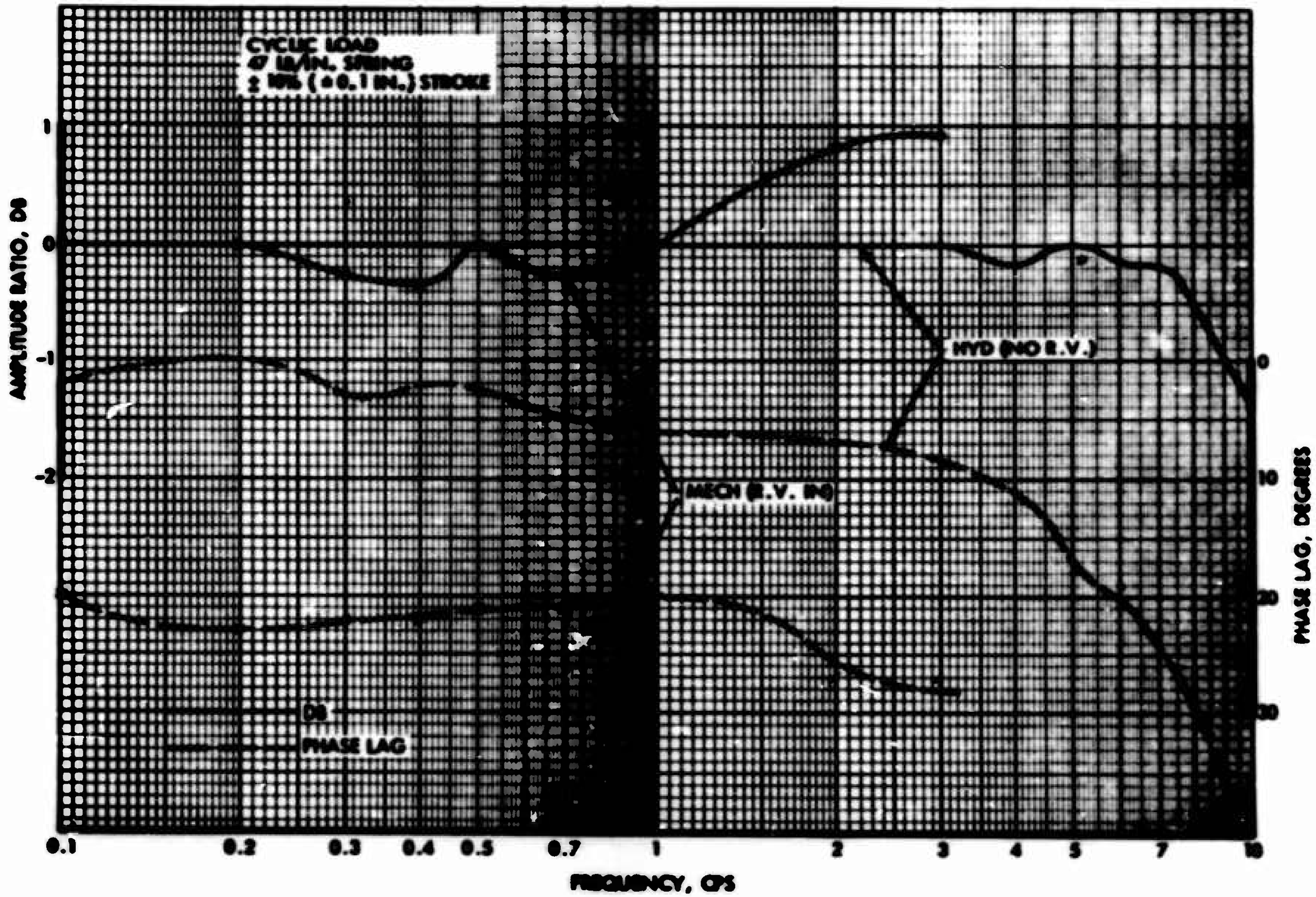

FIGURE 10 FREQUENCY RESPONSE, CYCLIC LOAD, 47-POUND SPRING, 10-PERCENT STROKE

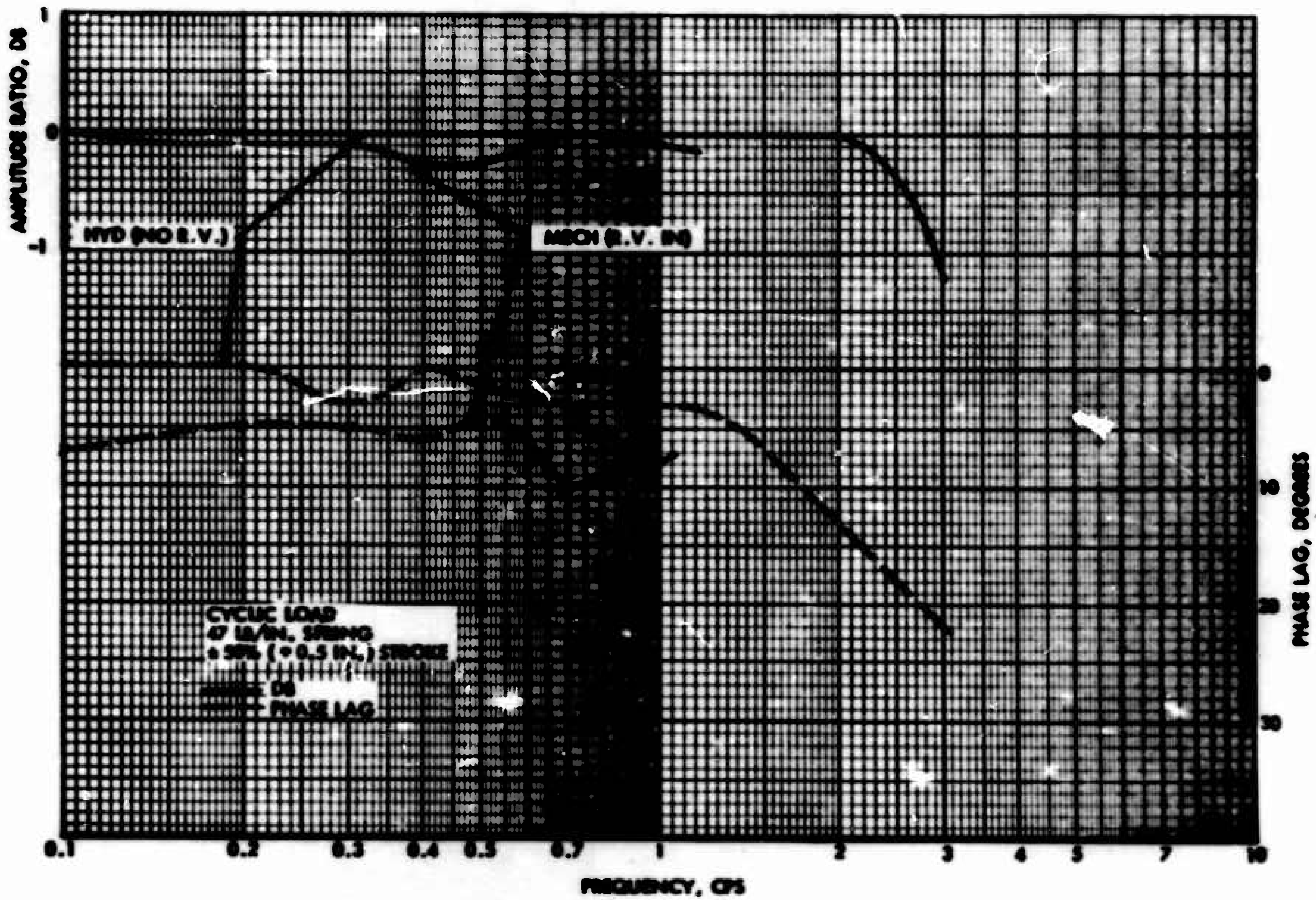

FIGURE 11 FREQUENCY RESPONSE, CYCLIC LOAD, 47-POUND SPRING, 50-PERCENT STROKE

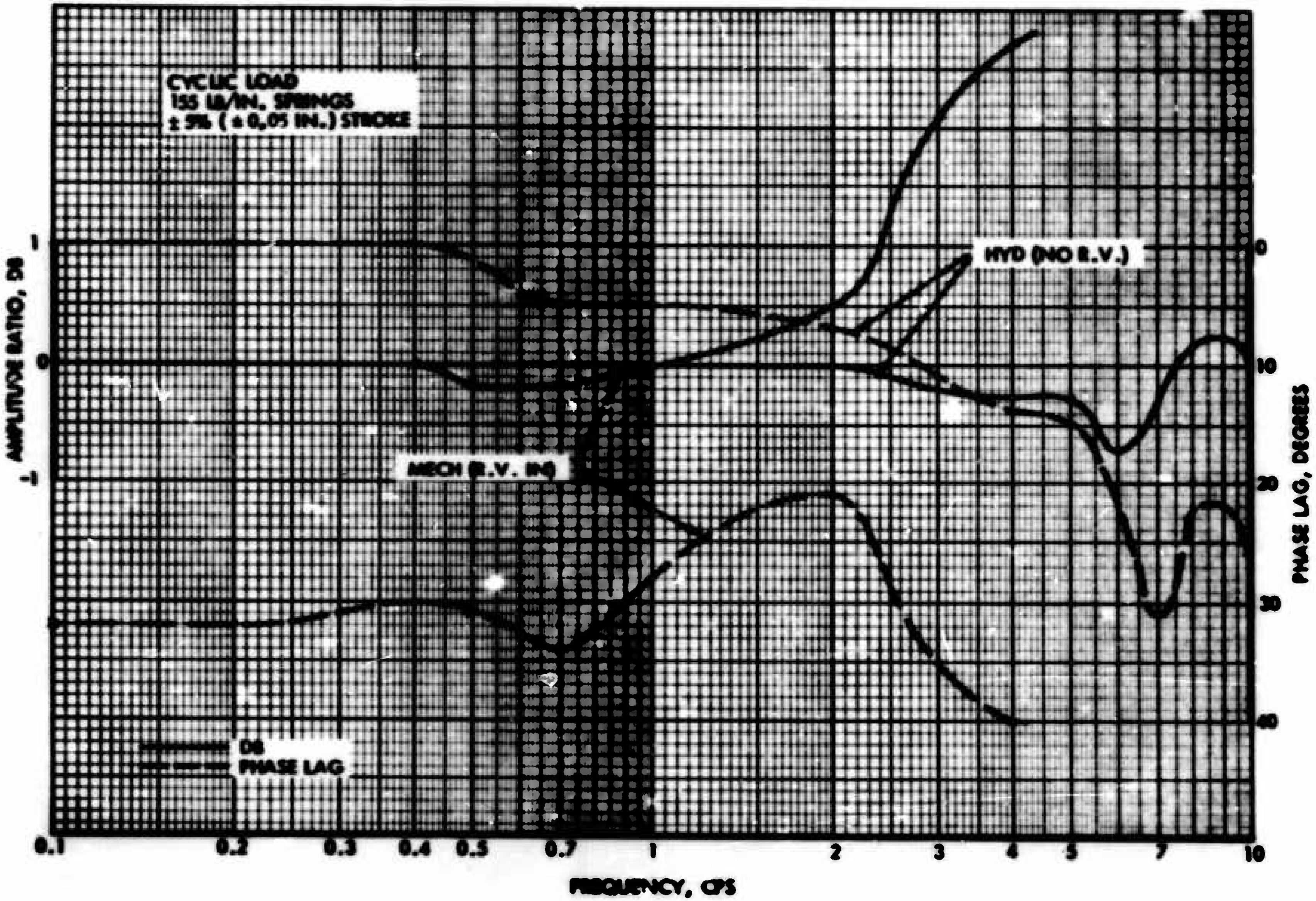

FIGURE 12 FREQUENCY RESPONSE, CYCLIC LOAD, 155-POUND SPRING, 5-PERCENT STROKE

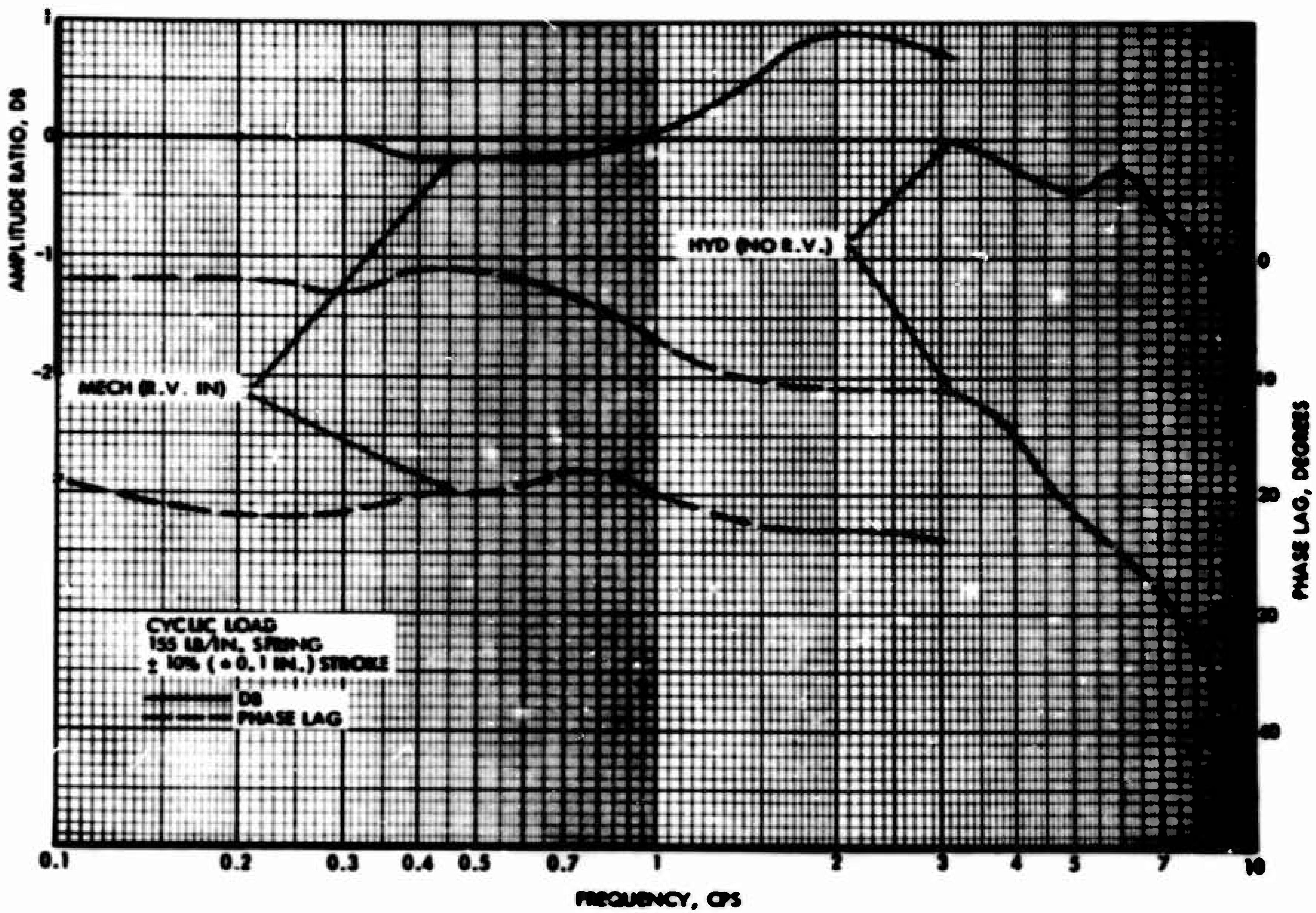

FIGURE 13 FREQUENCY RESPONSE, CYCLIC LOAD, 155-POUND SPRING, 1C-PERCENT STROKE

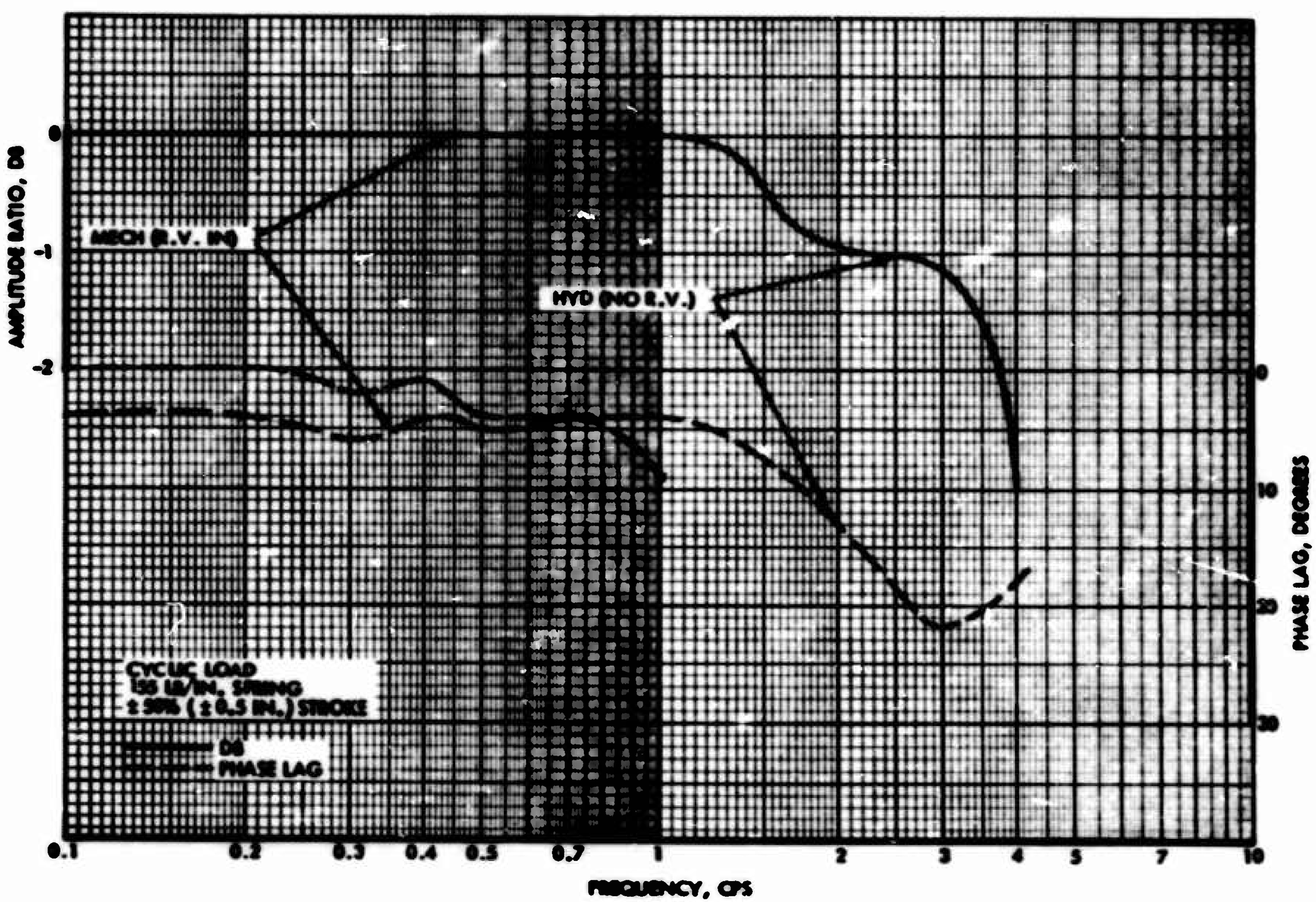

FIGURE 14 FREQUENCY RESPONSE, CYCLIC LOAD, 155-POUND SPRING, 50-PERCENT STROKE

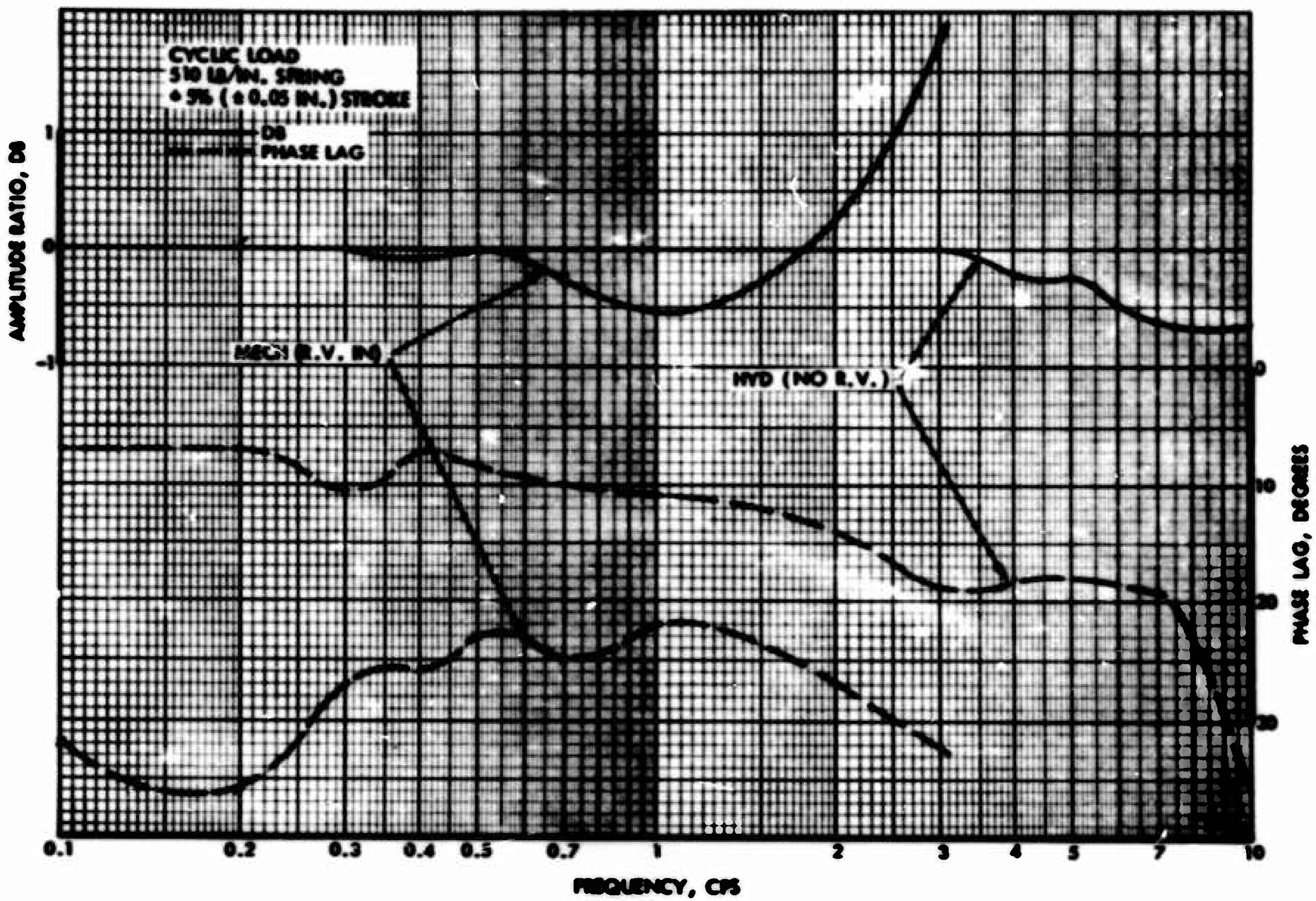

FIGURE 15 FREQUENCY RESPONSE, CYCLIC LOAD, 510-POUND SPRING, 5-PERCENT STROKE

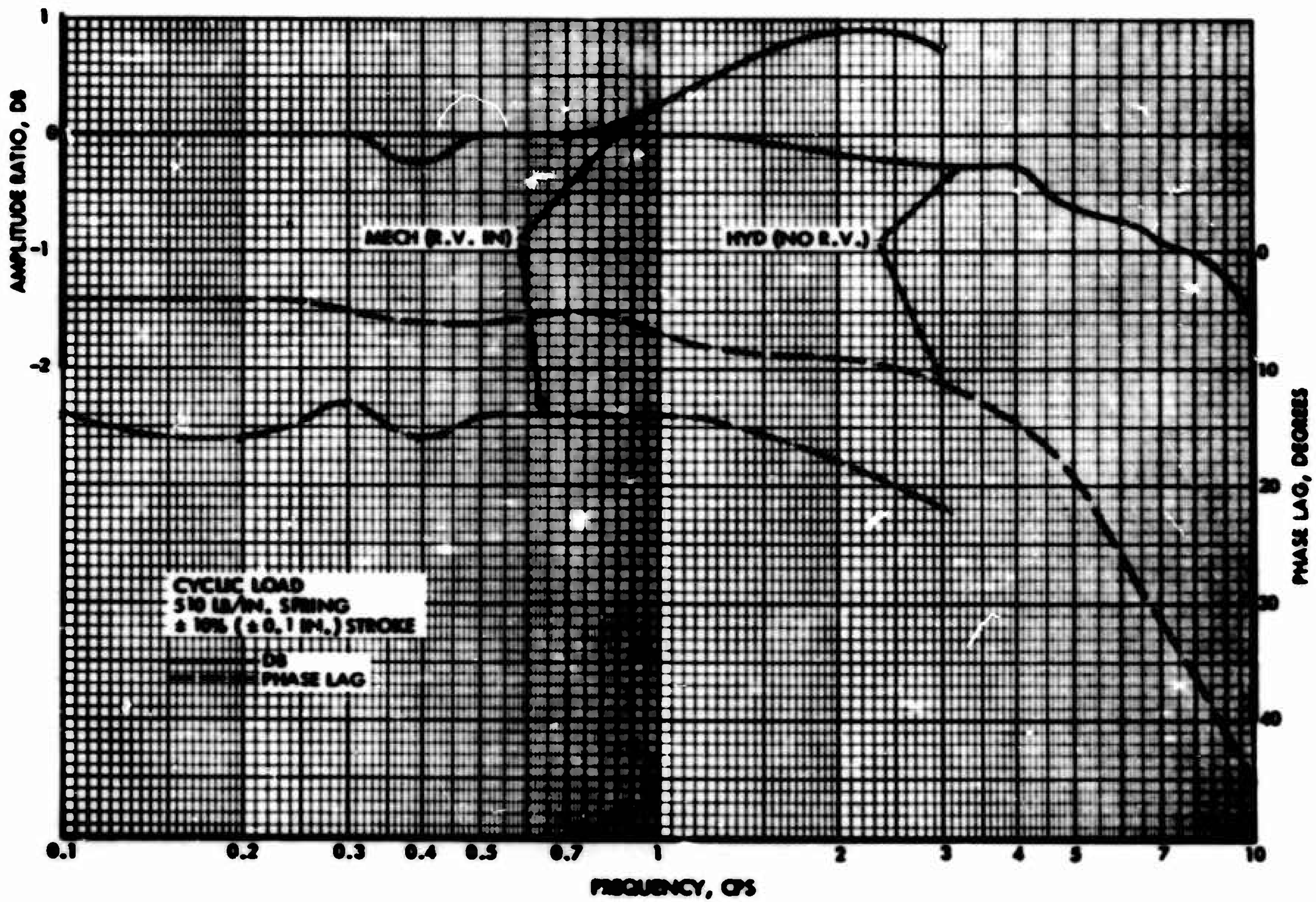

FIGURE 16 FREQUENCY RESPONSE, CYCLIC LOAD, 510-POUND SPRING, 10-PERCENT STROKE

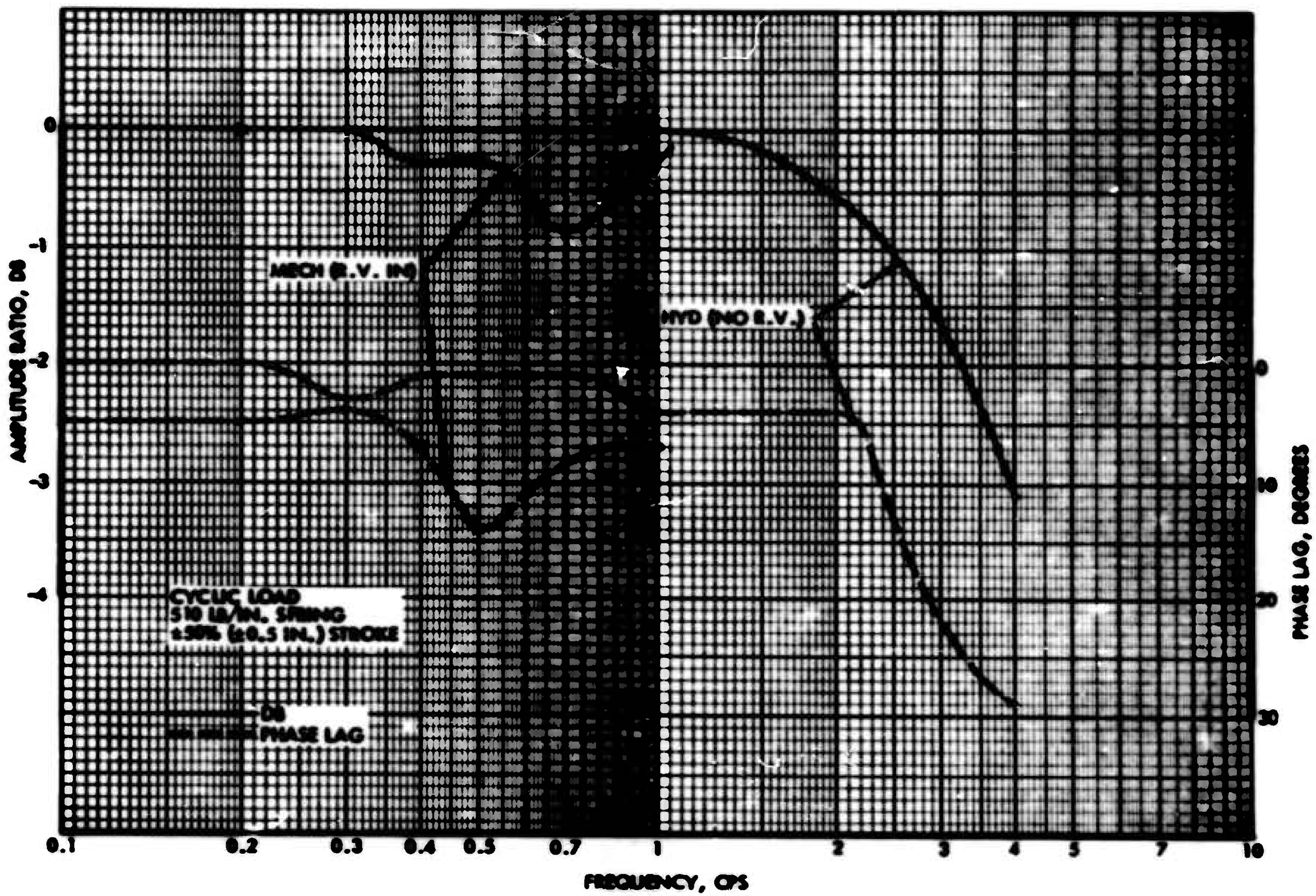

FIGURE 17 FREQUENCY RESPONSE, CYCLIC LOAD, 510-POUND SPRING, 50-PERCENT STROKE

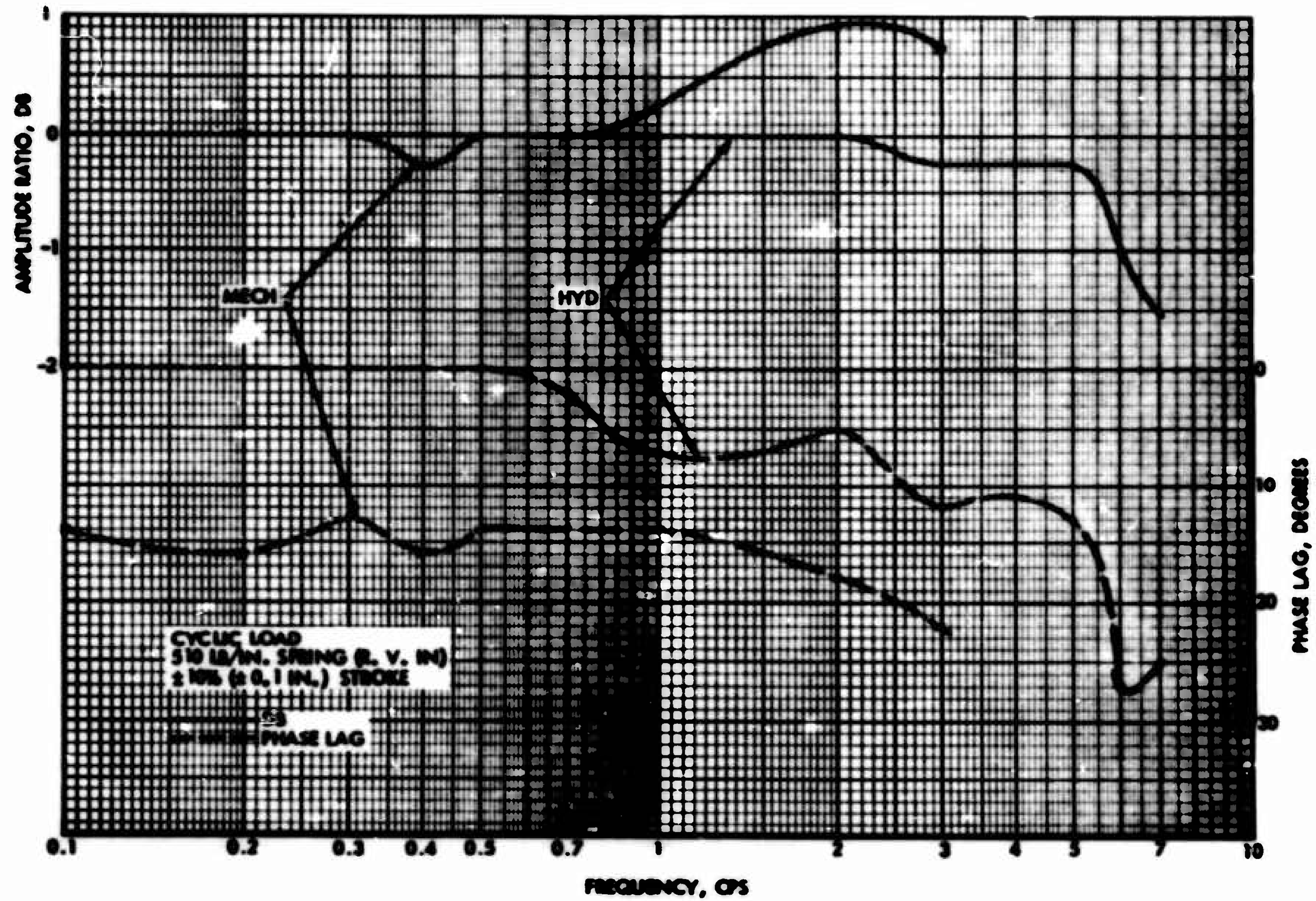

FIGURE 18 FREQUENCY RESPONSE, CYCLIC LOAD, 510-POUND SPRING, 10-PERCENT STROKE, RELIEF VALVE IN

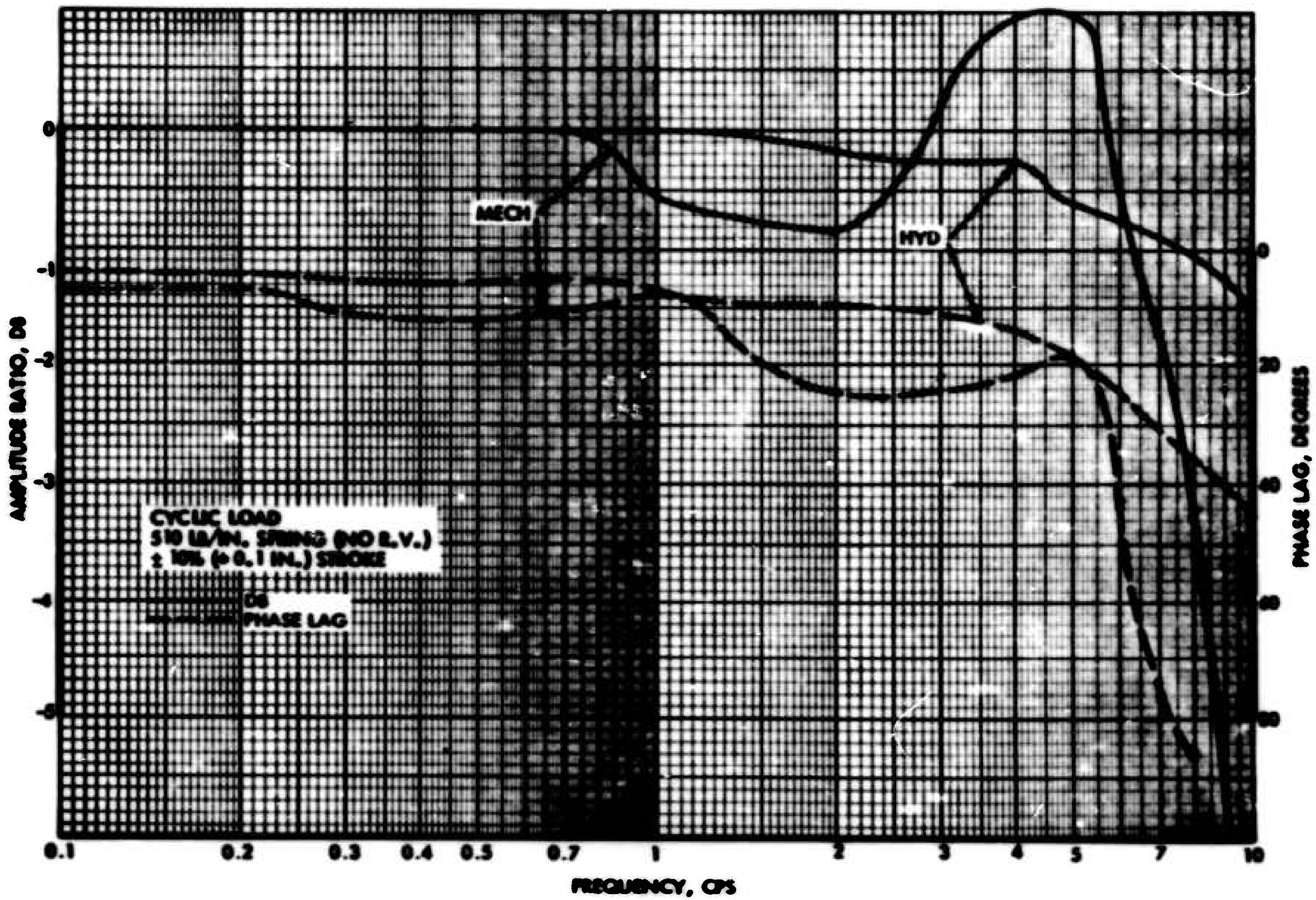

FIGURE 19 FREQUENCY RESPONSE, CYCLIC LOAD, 510-POUND SPRING, 10-PERCENT STROKE, NO RELIEF VALVE

TABLE 2

RESOLUTION TEST RESULTS

Output Direction	Arm Stroke Inch	Direction of Input	Minimum Input Motion Req'd, in.	Output Arm Motion, in.
		Cyclic - 47-Pound/Inch Spring		
Neutral	0	CW	.0005	Smooth
		CCW	.0003	Smooth
CW	.25	CW	.0005	.001 steps
		CCW	.0015	Smooth
CW	.50	CW	.001	.002 steps
		CCW	.002	.003 steps
CW	1.00	CW	.0005	Smooth
		CCW	.004	.008 steps
CCW	.25	CW	.0007	Smooth
		CCW	.0003	Smooth
CCW	.50	CW	.0007	.001 steps
		CCW	.001	Smooth
CCW	1.00	CW	.001	.003 steps
		CCW	.0007	Smooth
		155-Pound/Inch Spring		
Neutral	0	CW	.0005	Smooth
		CCW	.0005	Smooth
CW	.25	CW	.001	Smooth
		CCW	.004	.007 steps
CW	.50	CW	.001	.005 steps
		CCW	.004	.010 steps
CW	1.00	CW	.002	.004 steps
		CCW	Chattering, cannot measure	
CCW	.25	CW	.0015	.002 steps
		CCW	.001	Smooth
CCW	.50	CW	.002	.004 steps
		CCW	.0035	.008 steps
CCW	1.00	CW	.004	.008 steps
			Chattering, cannot measure	
		510-Pound/Inch Spring		
Neutral	0	CW	.001	Smooth
		CCW	.001	Smooth
CW	.25	CW	.002	.002 steps
		CCW	.014	.015 steps
CW	.50	CW	.009	Chattering, cannot measure
		CCW	.020	.030 steps
CW	1.00	CW	Chattering, cannot measure	
		CCW	.010	.045 steps
CCW	.25	CW	.0035	.007 steps
		CCW	.003 Approx.	Chattering
CCW	[illegible]	CW	.005	.012 steps
		CCW	.003 Approx.	Chattering

TABLE 2 (Cont.)

Output Direction	Arm Stroke Inch	Direction of Input	Minimum Input Motion Req'd, in.	Output Arm Motion, in.
CCW	1.00	CW	Overshooting motion, cannot measure*	
		CCW	Chattering, cannot measure	
	Collective Load - 10-Pound/Inch Spring and 20-Pound Mass			
CW	.01	CW	.0005	Smooth
		CCW	.001	Smooth
CW	1.00	CW	.001	Tends to keep moving
		CCW	.0015	.008 steps
CW	2.00	CW	.001	Tends to keep moving
		CCW	.0015	.010 steps
CW	3.90	CW	.008	Tends to keep moving
		CCW	.004	.020 steps
		50-Pound/Inch Spring and 20-Pound Mass		
CW	.01	CW	.0005	Smooth
		CCW	.0025	Smooth
CW	1.00	CW	.0008	Smooth
		CCW	.0015	.020 steps
CW	2.00	CW	.001	Chattering cannot measure
		CCW	Overshoot motion, cannot measure	
CW	3.90	CW	Chattering, cannot measure	
		CCW	Overshooting motion, cannot measure	
	After Endurance Cycling Test - Cyclic - 510-Pound/Inch Spring			
CW	.01	CW	.002	Smooth
		CCW	.001	Smooth
CW	.25	CW	.002	.003 steps
		CCW	.005	.007 steps
CW	.50	CW	.003	.006 steps
		CCW	.018	.040 steps
CW	1.00	CW	.005	.005 steps
		CCW	Overshooting motion, cannot measure	
CCW	.25	CW	.006	.010 steps
		CCW	.001	.004 steps
CCW	.50	CW	.005-.020	.015-.050 steps
		CCW	.005	Chattering cannot measure
CCW	1.00	CW	Overshooting motion, cannot measure	
		CCW	Chattering, cannot measure	
	Collective Load - 10-Pound/Inch Spring and 20-Pound Mass			
CW	2.00	CW	.001	Smooth
		CCW	.002	.004 steps

*Was unable to move input arm to edge of dead-band area without the output arm moving sufficiently to cause the input arm to be in the dead-band area again.

TABLE 3

MAXIMUM RATE TEST RESULTS

Servo-Actuator	Type Load	Load Condition	Output Motion	Total Stroke, In.	Load Rate,* In/Sec
Mech. (R.V. installed)	Cyclic	47-lb/in. Spring	CW	1.90	2.00
			CCW		2.54
Mech. (After Endurance Test) (Manual Input)	Cyclic	510-lb/in. Spring	CW	1.76	2.50-4.16
			CCW		1.59-2.78
Hyd.	Cyclic	47-lb/in. Spring	Retract	1.75	3.16
			Extend		3.13
(Manual Input)		155-lb/in. Spring	Retract	1.70	3.29
			Extend		3.33
		510-lb/in. Spring	Retract	1.70	2.73
			Extend		2.7[illegible]
Mech.	Collective	(10-lb/in. Spring)	CW	3.96	4.52
		(20-lb Mass)	CCW		6.62
(R.V. installed)		(10-lb/in. Spring)	CW		4.62
		(40-lb Mass)	CCW		6.62
		(50-lb/in. Spring)	CW		4.22
		(20-lb Mass)	CCW		6.84
		(50-lb/in. Spring)	CW		4.22
		(40-lb Mass)	CCW		7.34
Mech. (After Endurance Test)	Collective	(10-lb/in. Spring)	CW		6.20
		(20-lb Mass)	CCW		11.7
Hyd.	Collective	(10-lb/in. Spring)	Retract	3.95	3.35
		(20-lb Mass)	Extend		3.53
(R.V. installed)		(10-lb/in. Spring)	Retract		3.29
		(40-lb Mass)	Extend		3.41
		(50-lb/in. Spring)	Retract		3.19
		(20-lb Mass)	Extend		4.29
		(50-lb/in. Spring)	Retract		3.14
		(20-lb Mass)	Extend		4.29

*The motion of the output load arm bell crank was recorded on a Sanborn recorder and the maximum rates were determined by measuring the slopes of the recorded traces.

R.V. = Relief Valve

TABLE 4

STEP RESPONSE TEST RESULTS

Load Conditions	Step	Stroke, Inches	Output Motion	Time Constant, Sec Mech.	Hyd.
		Cyclic Condition			
47-Pound/Inch Spring Mech. (R.V. In)	10%	0.1	CW/Ret. CCW	.013 .022	.010
Hyd. (No R.V.)	25%	.25	CW/Ret. CCW	.006 .011	.012
	50%	.50	CW/Ret. CCW	.010 .020	.020
115-Pound/Inch Spring Mech. (R.V. In)	10%	.1	CW/Ret. CCW	.015 .015	.008
Hyd. (No R.V.)	25%	.25	CW/Ret. CCW	.007 .022	.018
	50%	.50	CW/Ret. CCW	.007 .031	.020
510-Pound/Inch Spring Mech. (R.V. In)	10%	.1	CW/Ret. CCW	(-).008 .030	.013
Hyd. (No R.V.)	25%	.25	CW/Ret. CCW	(-).002 .010	.018
	50%	.50	CW/Ret. CCW	.010 .010	.019
510-Pound/Inch Spring (No R.V.)	10%	.1	CW CCW	.012 .025	
(After Endurance Test)	25%	.25	CW CCW	.025 .042	
10-Pound/Inch Spring 20-Pound Mass	10%	.4	CW/Ret. CCW	.005 .050	.018 .018
	25%	1.0	CW/Ret. CCW/Ext.	.010 .040	.028 .021
	50%	2.0	CW/Ret. CCW/Ext.	.025 .005	.032 .015
Mech. (No R.V.) (After Endurance Test)	10%	.4	CW CCW	.022 .028	
10-Pound/Inch Spring 40-Pound Mass	10%	.4	CW/Ret. CCW/Ext.	.015 .055	.005 .005

TABLE 4 (Cont.)

Load Conditions	Step	Stroke, Inches	Output Motion	Time Constant, Sec Mech.	Hyd.
		Cyclic Condition			
	25%	1.0	CW/Ret.	(-).030	.030
			CCW/Ext.	.030	.028
	50%	2.0	CW/Ret.	.015	.040
			CCW/Ext.	.015	.015
50-Pound/Inch Spring 20-Pound Mass	10%	.4	CW/Ret.	(-).010	.012
			CCW/Ext.	.025	.010
	25%	1.0	CW/Ret.	(-).005	.022
			CCW/Ext.	.025	.017
	50%	2.0	CW/Ret.	.015	.032
			CCW/Ext.	.020	.018
50-Pound/Inch Spring	10%	.4	CW/Ret.	.005	.028
			CCW/Ext.	.030	.015
	25%	1.0	CW/Ret.	(-).010	.020
			CCW/Ext.	.010	.015
	50%	2.0	CW/Ret.	.030	.040
			CCW/Ext.	.015	.018

NOTES:
- CW - Clockwise
- CCW - Counterclockwise
- Ret. - Retraction
- Ext. - Extension
- (-) - Denotes that output rate of mechanical servo-actuator was faster than the rate of the input hydraulic cylinder.
- R.V. - Relief Valve

TABLE 5

ENDURANCE CYCLING TEST

Cycling Group No.	No. of Cycles	Output Stroke Inch	Freq. CPS	Remarks
1-a	200	2.0	.1	Actuator housing surface reached +175°F at end of cycling period. Noted chattering noise on the counter-clockwise rotation of the actuator. Intermittent noise on clockwise end of load - possibly bearing noise.
1-b	1000	1.0	.5	Replaced input power flexible shaft during this run. Housing stayed below +120°F during run.
1-c	4000	.05	5.0	Functioned normally.
2-a	200	2.0	.1	Output CCW stroke started to become shorter after approximately 14.0 cycles. Housing temperature was +157°F. Removed from test fixture and disassembled. Found bonded joint of Delrin gear (P/N 173428) and hub (P/N 173427) has failed allowing Delrin gear to come out of mesh. Repaired joint by adding a locking pin between the hub and pinion plus 1/4 in. washer to prevent pin from coming out. Next, riveted Delrin gear to hub with 4 aluminum rivets. Reassembled unit and remounted in test fixture. Still unable to cycle actuator properly. It was disassembled again, but could not find any reason for malfunctioning. Decided to reassemble and continue cycling with the lower amplitudes only.
2-b	1000	1.0	.5	Unit chatters but able to obtain full amplitude. Housing surface temperature reached a maximum of +127°F.
2-c	4000	.05	5.0	Functioned normally.
3-b	1000	1.0	.5	Functioned the same as before - full stroke but chatters. Housing surface temperature reached +143°F.
3-c	4000	.05	5.0	Functioned normally.
4-b	1000	1.0	.5	No change, although oil is starting to leak out of the front seal - seeps only.
4-c	4000	.05	5.0	Functioned normally.

TABLE 5 (Cont.)

Cycling Group No.	No. of Cycles	Output Stroke Inch	Freq. CPS	Remarks
5-b	1000	1.0	.5	No change. Housing temperature increased from +140°F to +125°F during this period of cycling.
5-c	4000	.05	5.0	Functioned normally. At completion of this step all of the required cycling except for the full 2.00-inch cycling was complete. Started to measure force threshold and it was noted to be high. The actuator was again disassembled and it was found that the #70 dowel pin on Print 173410 had worked out. This pin is between gear (P/N 173421) and bushing (P/N 173422). The pin showed evidence of rubbing against the metal flange of Delrin gear (173419). Reset the pin and reassembled actuator. It was decided to continue with the full stroke cycle.
2-a	60	2.0	.1	Completed remainder of Group 2-a cycles.
3-a	200	2.0	.1	Housing temperature increased from +118°F to +184°F during the run. CCW stroke has started shortening again.
4-a	200	2.0	.1	Allowed unit to cool to +99°F before starting this phase. Still has restricted motion in CCW direction. Disassembled the actuator again and fround #70 dowel pin had backed out again. Put a second retaining pin in gear shaft and #70 dowel pin. Reassembled the actuator and completed this phase of cycling. Housing temperature had reached +155°F at end of this phase.
5-a	200	2.0	.1	Housing surface temperature has increased from +116°F to +170°F during the phase. Actuator still has a restricted motion in the CCW direction. Sounds like something inside of actuator is hitting mechanical stop. Input force jumps to approximately 200 lbs. when the CCW motion is restricted. Figure 33 shows a Sanborn trace of the actuator cycling. At completion of cycling, the unit was again disassembled; however, could not find evidence of binding between the input arm and the spring bungee. Noted some wear marks on the brass bushing opposite the loading gear.

that in each set of tests the maximum rates were greater when the collective spring load was aiding (counterclockwise direction).

Collective Load - Frequency Response Test

The mechanical servo-actuator was cycled sinusoidally at the output amplitudes of ±0.40, ±0.80 and ±1.8 inches. The frequency was varied from 0.1 cps to a maximum of 4 cps (relief valves limiting higher frequencies) for each of the four loading conditions. After completion of endurance cycling, the frequency response test was repeated using the 10-pound/inch spring and the 20-pound mass attached to the output arm of the mechanical servo-actuator. The hydraulic relief valves were removed prior to this last test. Results of these tests are shown in Figures 20 through 32.

Collective Load - Step Response Test

The mechanical servo-actuator was operated at maximum rate with output steps of 0.40, 1.00, and 2.00 inches for all four collective loading conditions. After the endurance cycling test, the 10-pound/inch spring and 20-pound mass were attached to the output arm and the actuator was subjected to a step resonse test of 0.40-inch and 1.00-inch steps without the hydraulic relief valves installed. The time constant for each step was determined as described in the cyclic load section, and the results are tabulated in Table 4.

Endurance Cycling - Mechanical Servo-Actuator

After the above tests had been completed, the 510-pound/inch cyclic load spring was attached to the output arm of the mechanical servo-actuator and subjected to the following series of cycling tests:

1) 1,000 cycles at 100-percent amplitude and a frequency of 0.1 cps.

2) 5,000 cycles at 50-percent amplitude and a frequency of 0.5 cps.

3) 20,000 cycles at 5-percent amplitude and a frequency of 5 cps.

Each of the above groups of cycles was divided into five incremental portions. The actuator was cycled within each of the increments except as noted in Table 5. A sample of the Sanborn traces recorded during the endurance test is shown in Figure 33.

Emergency Operation Test

A loss of input mechanical power to the mechanical servo-actuator was simulated, thereby requiring a manual input force to operate the actuator with a 510-pound/inch cyclic load spring attached to the output arm of the actuator. The actuator was operated through a full stroke of 2.20 inches as the input hydraulic cylinder traveled through a full stroke of 1.78 inches. Figure 34 represents a Sanborn trace of the output load and

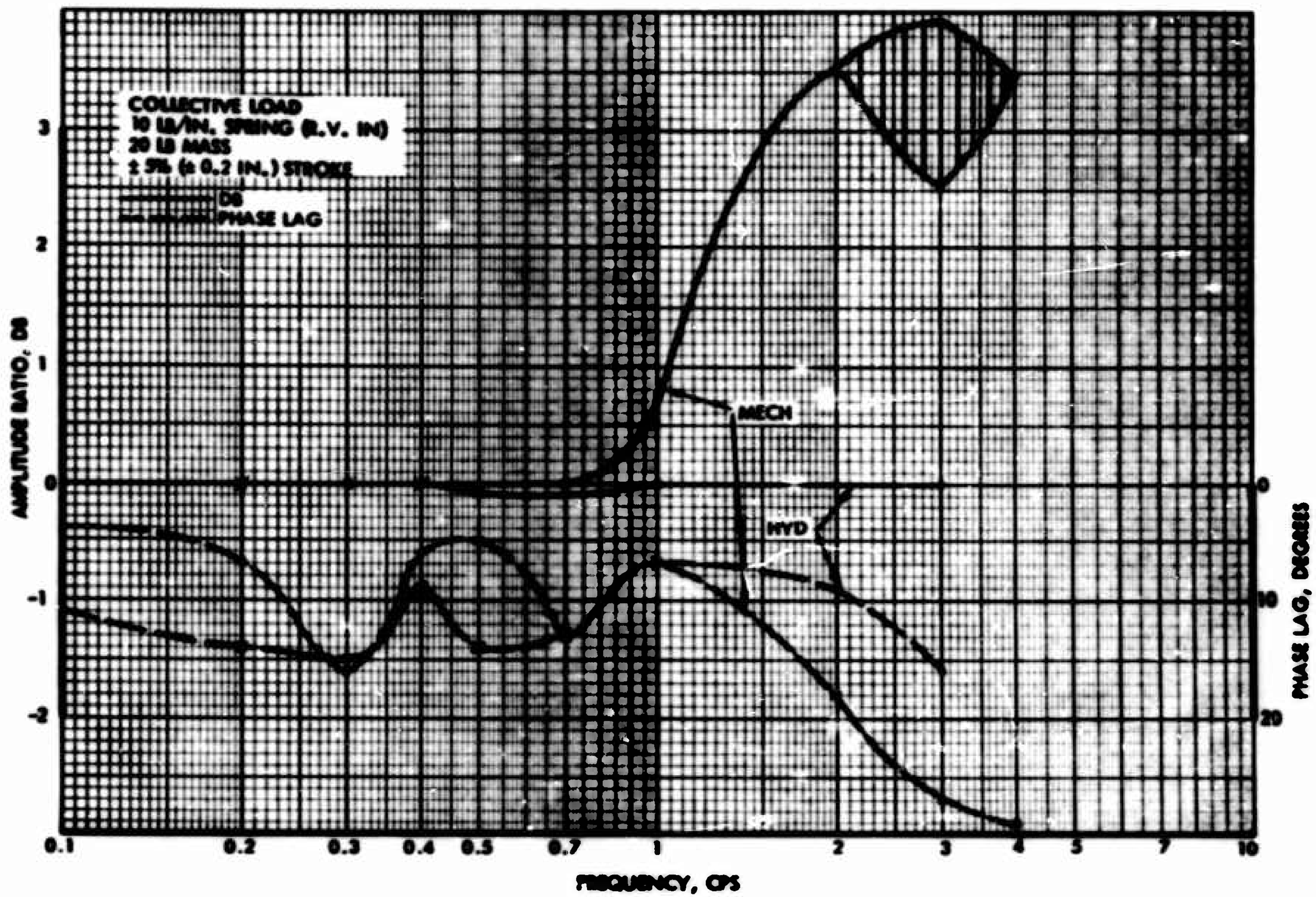

FIGURE 20 FREQUENCY RESPONSE, COLLECTIVE LOAD, 10-POUND SPRING, 20-POUND MASS, 5-PERCENT STROKE

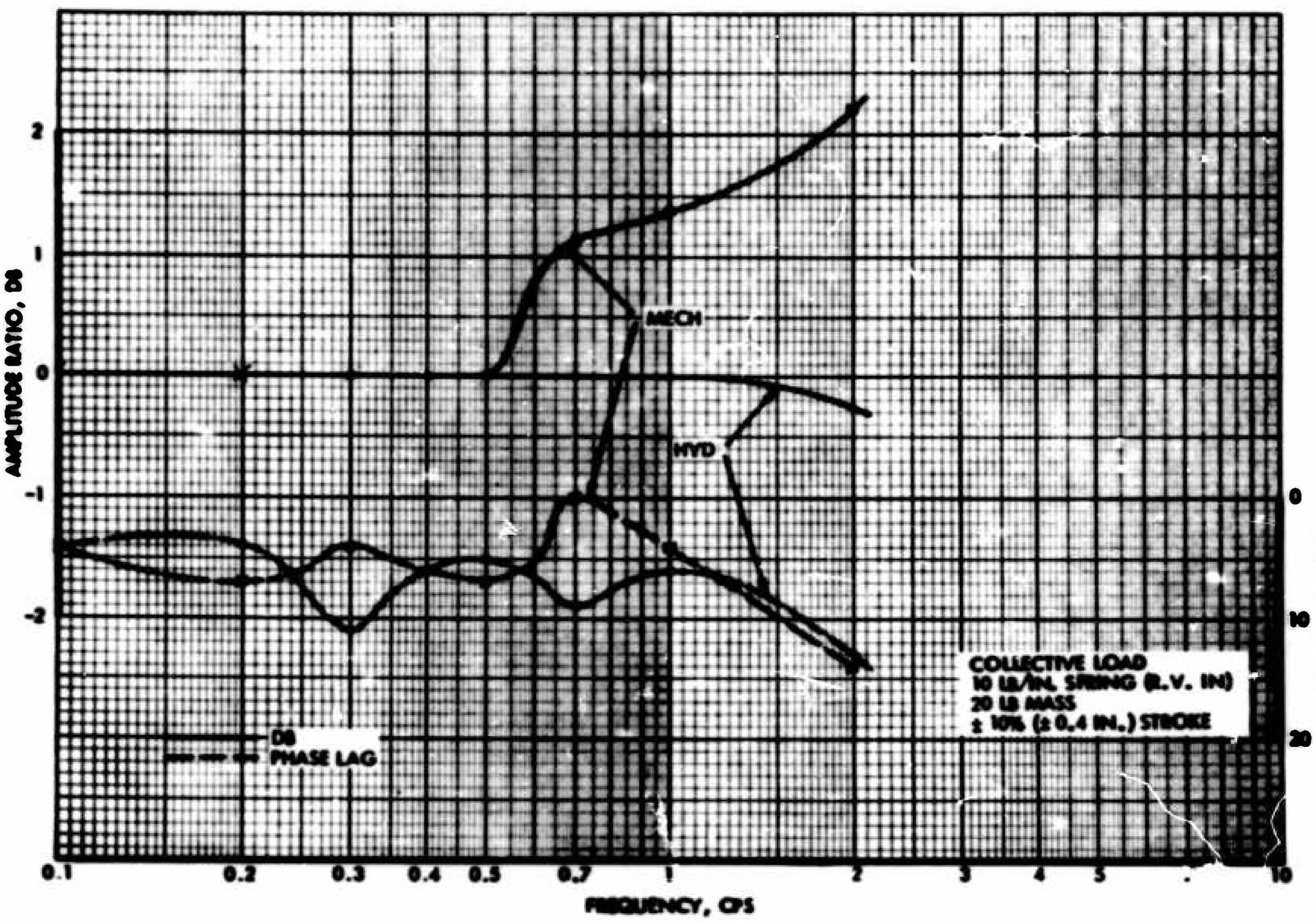

FIGURE 21 FREQUENCY RESPONSE, COLLECTIVE LOAD, 10-POUND SPRING, 20-POUND MASS, 10-PERCENT STROKE

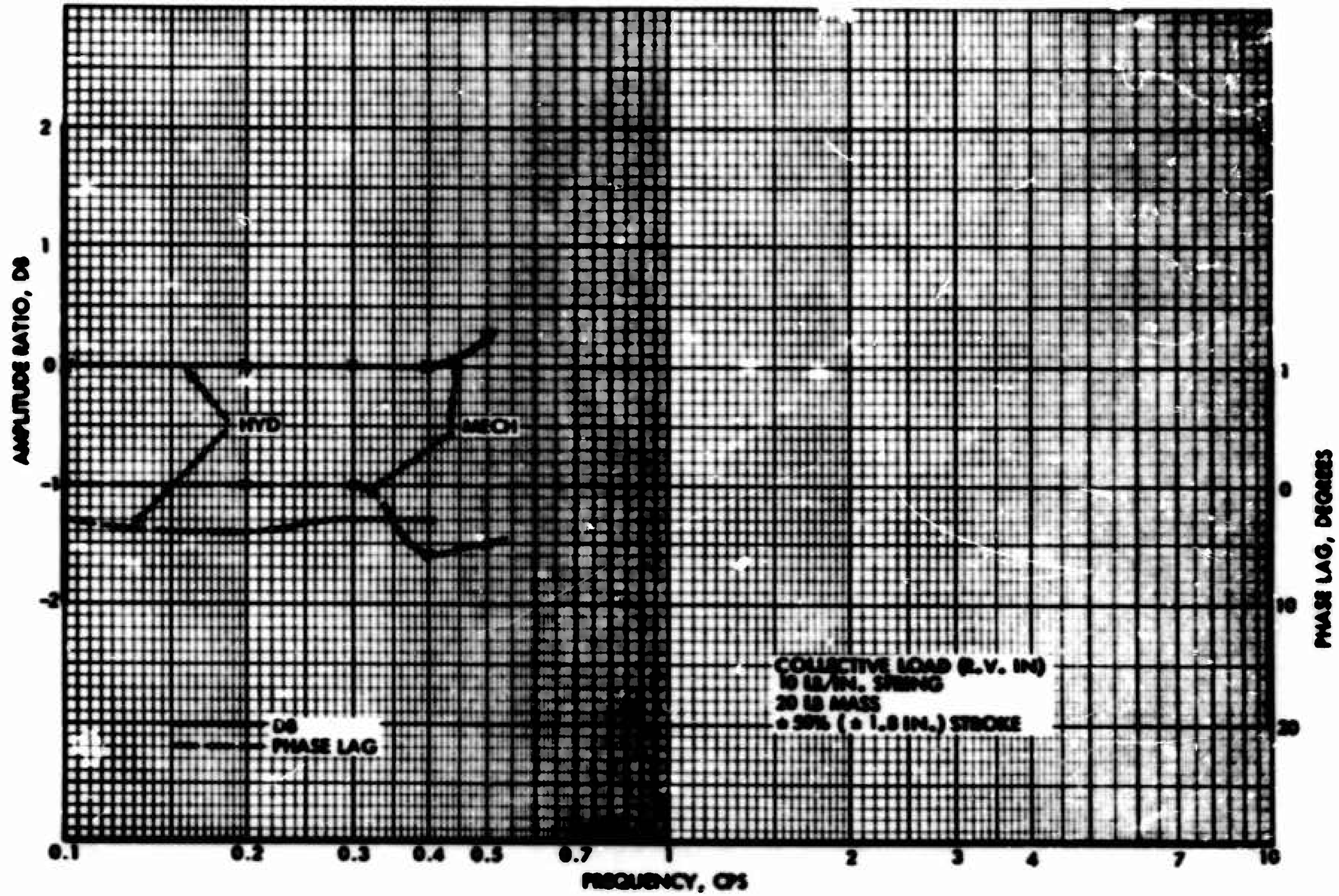

FIGURE 22 FREQUENCY RESPONSE, COLLECTIVE LOAD, 10-POUND SPRING, 20-POUND MASS, 50-PERCENT STROKE

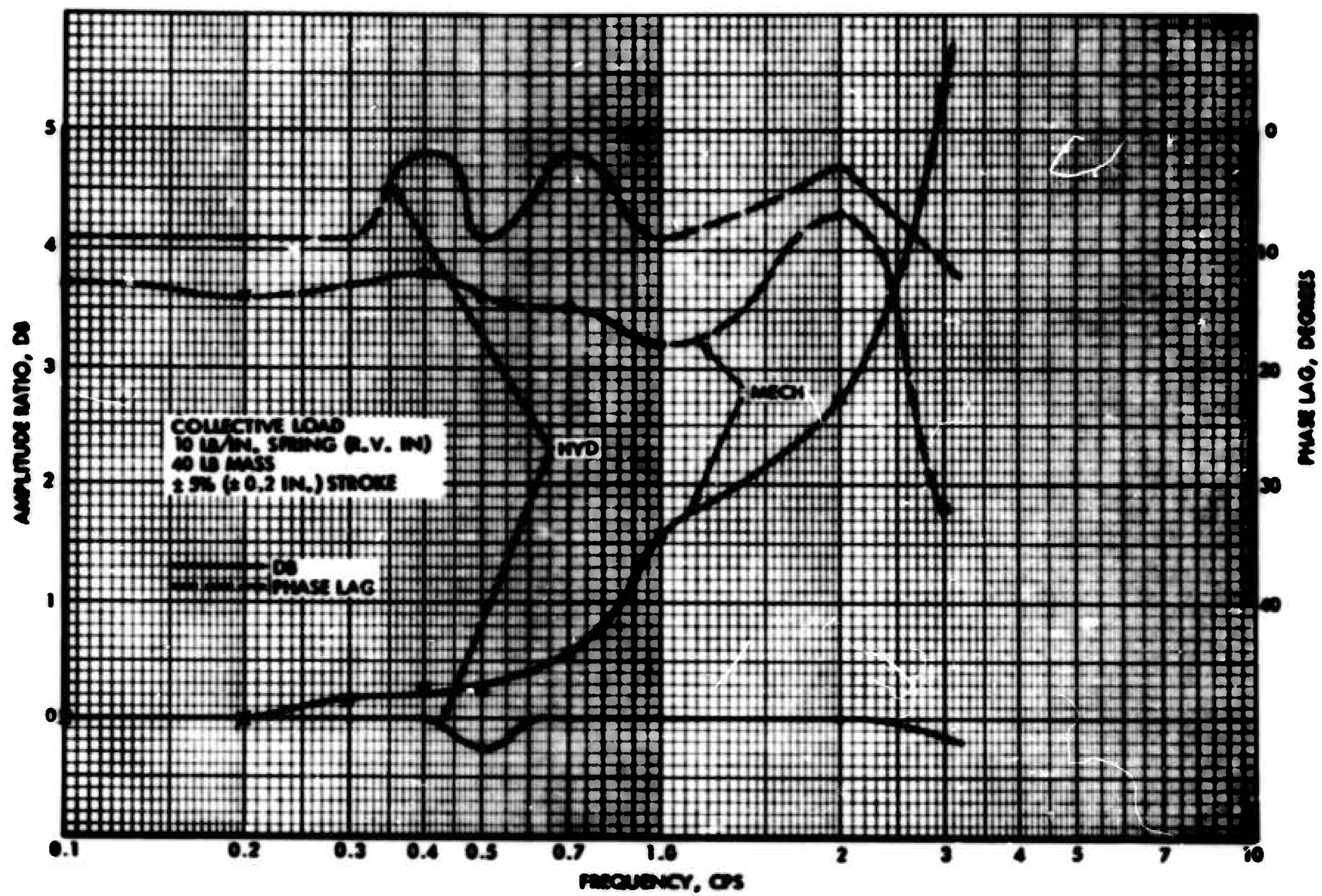

FIGURE 23 FREQUENCY RESPONSE, COLLECTIVE LOAD, 10-POUND SPRING, 40-POUND MASS, 5-PERCENT STROKE

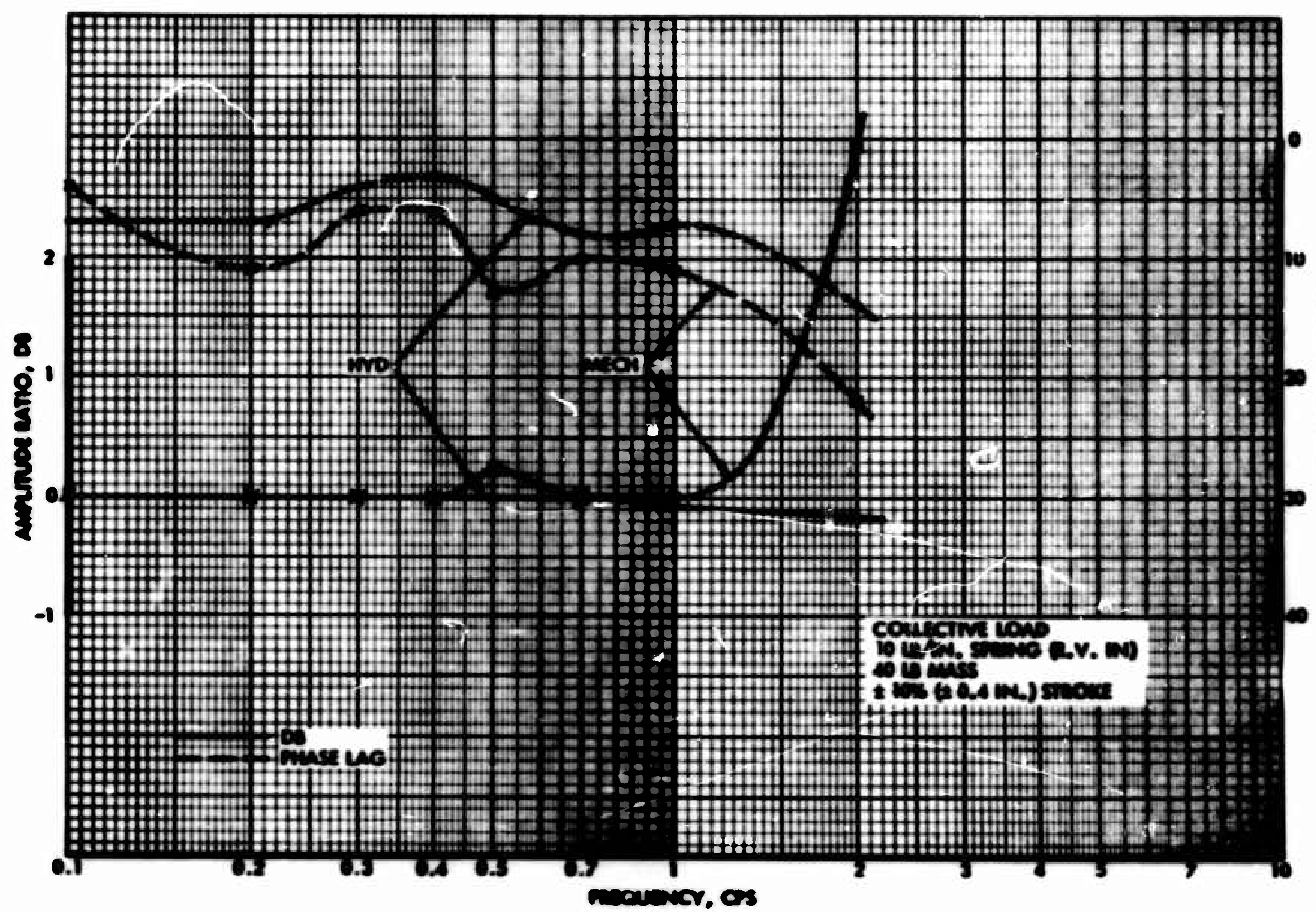

FIGURE 24 FREQUENCY RESPONSE, COLLECTIVE LOAD, 10-POUND SPRING, 40-POUND MASS, 10-PERCENT STROKE

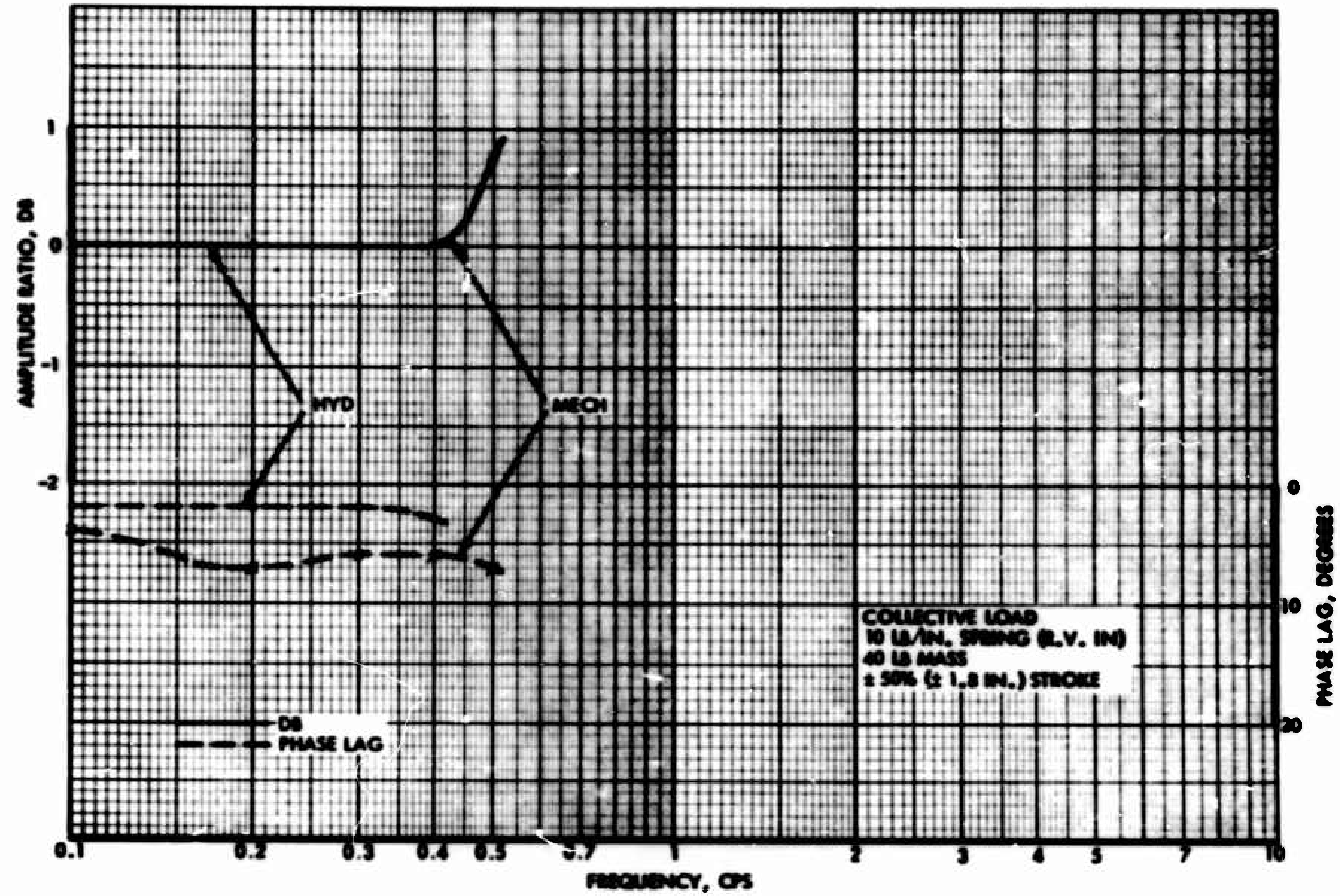

FIGURE 25 FREQUENCY RESPONSE, COLLECTIVE LOAD, 10-POUND SPRING, 40-POUND MASS, 50-PERCENT STROKE

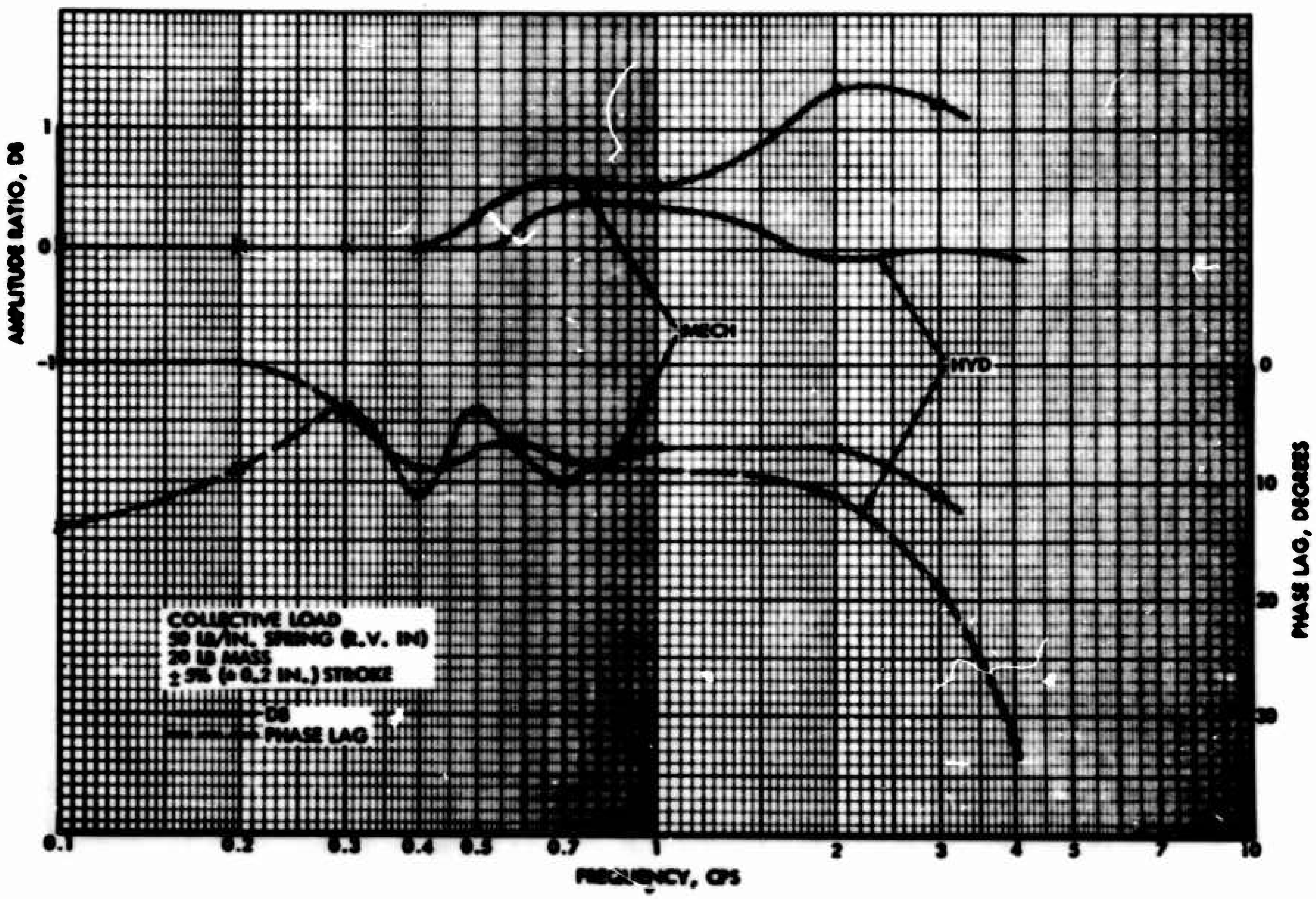

FIGURE 26 FREQUENCY RESPONSE, COLLECTIVE LOAD, 50-POUND SPRING, 20-POUND MASS, 5-PERCENT STROKE

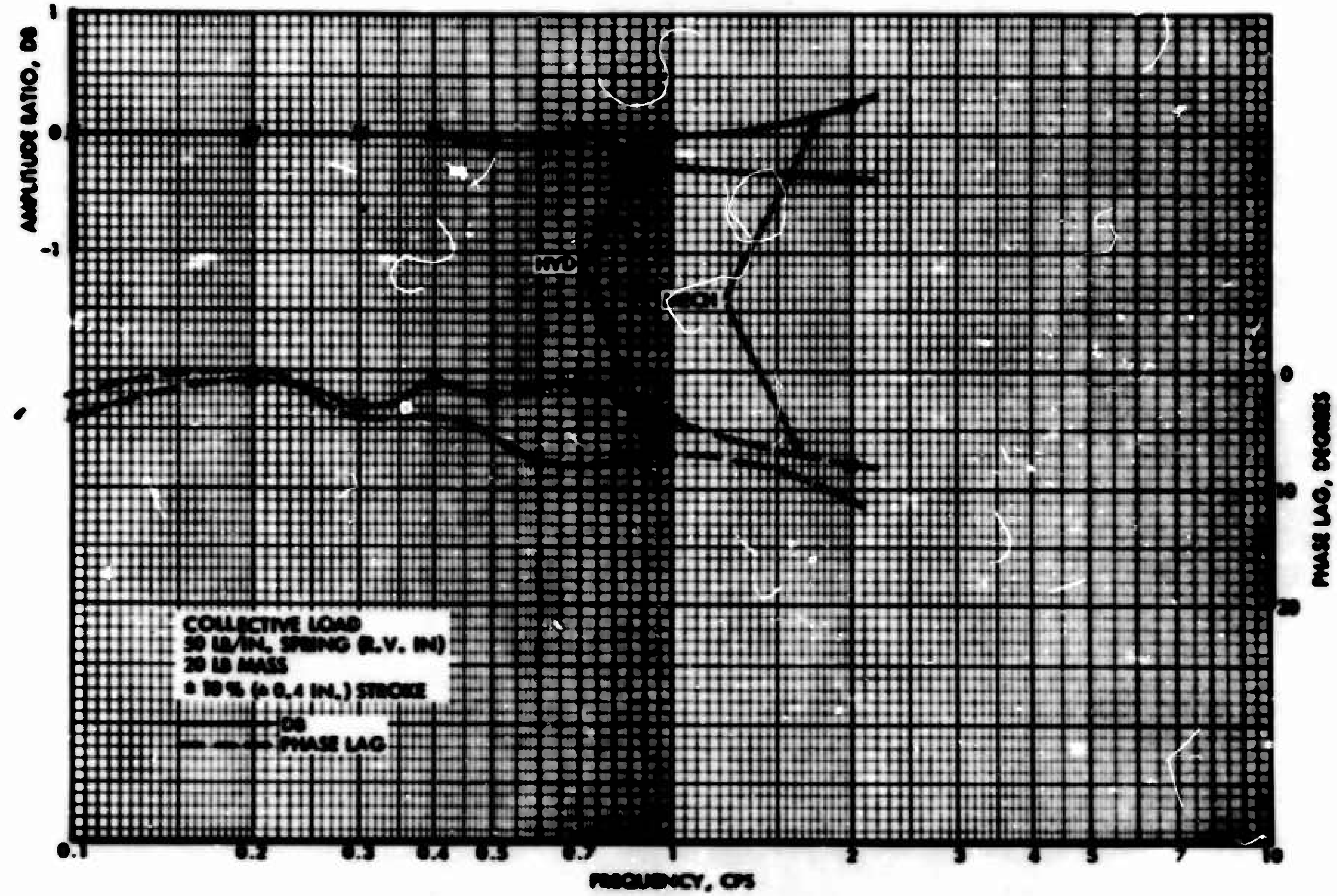

FIGURE 27 FREQUENCY RESPONSE, COLLECTIVE LOAD, 50-POUND SPRING, 20-POUND MASS, 10-PERCENT STROKE

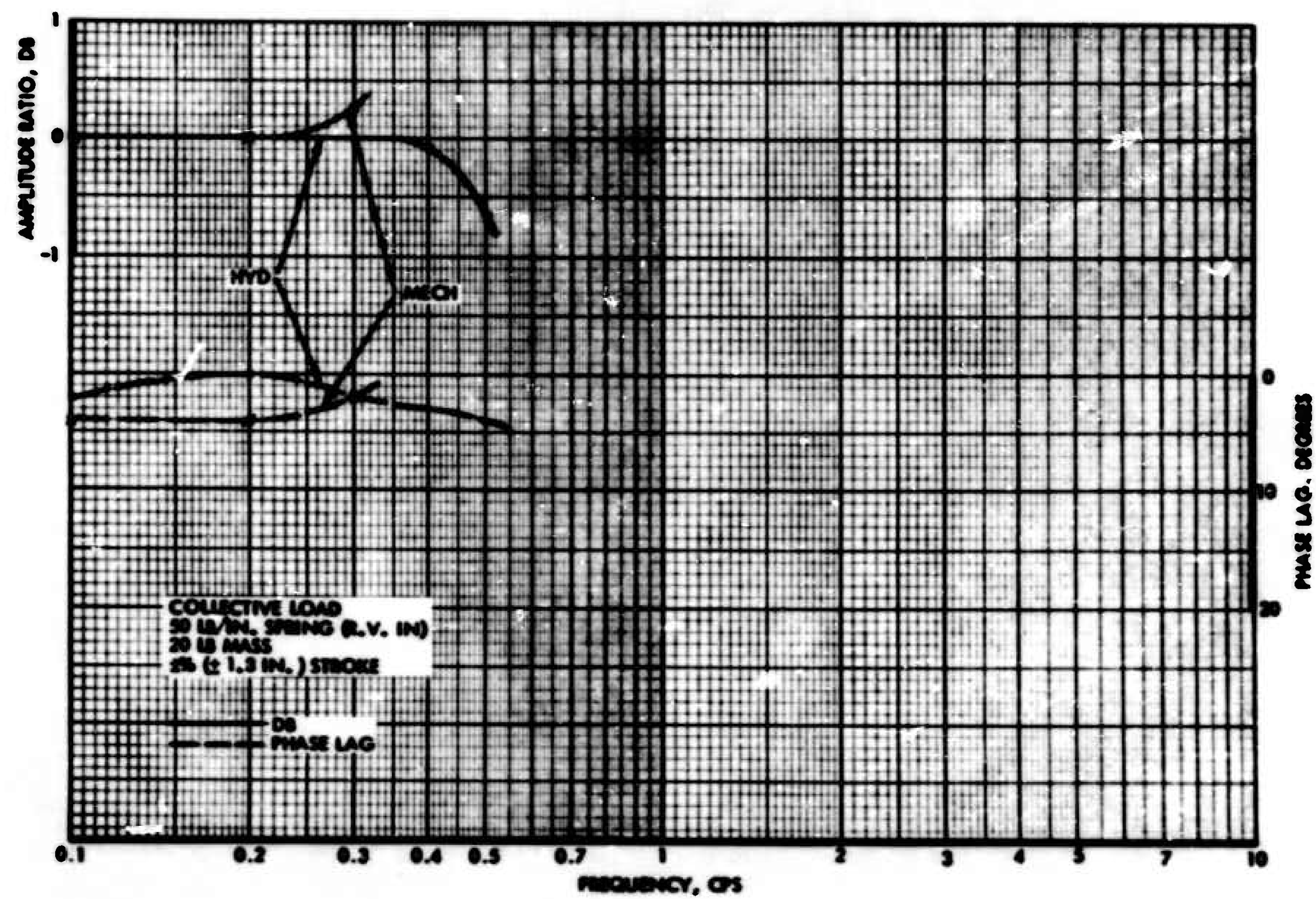

FIGURE 28 FREQUENCY RESPONSE, COLLECTIVE LOAD, 50-POUND SPRING, 20-POUND MASS, 50-PERCENT STROKE

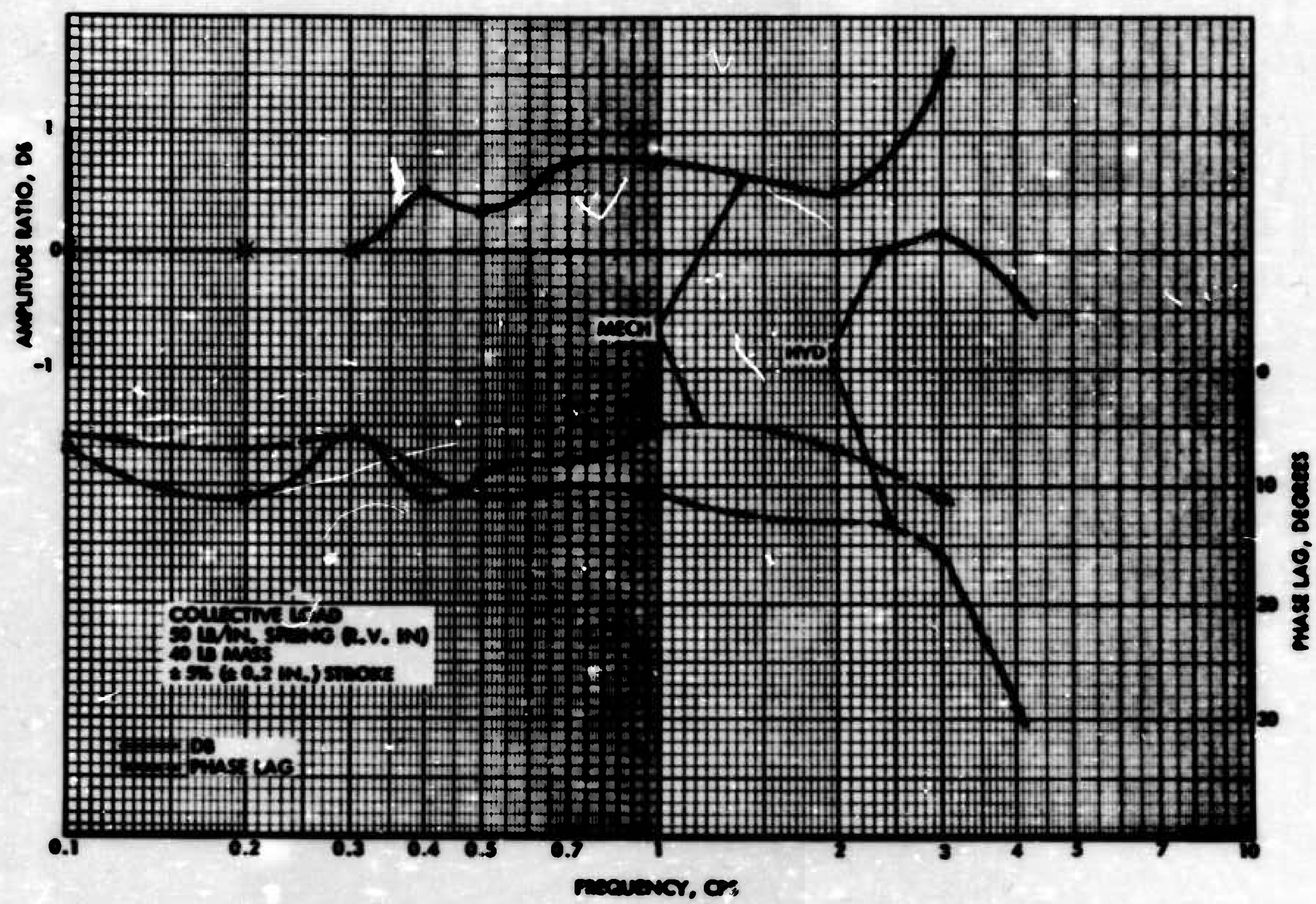

FIGURE 29 FREQUENCY RESPONSE, COLLECTIVE LOAD, 50-POUND SPRING, 40-POUND MASS, 5-PERCENT STROKE

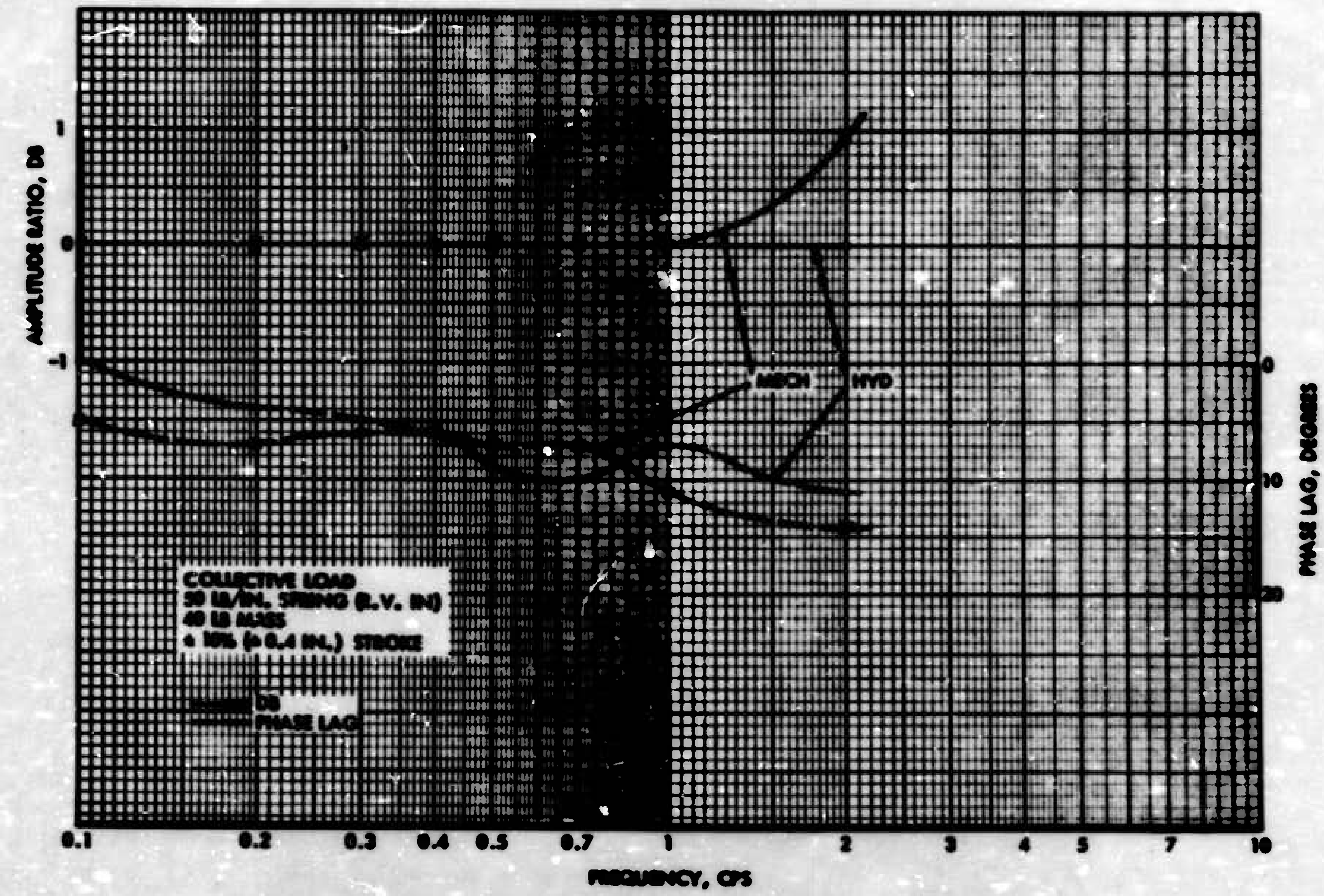

FIGURE 30 FREQUENCY RESPONSE, COLLECTIVE LOAD, 50-POUND SPRING, 40-POUND MASS, 10-PERCENT STROKE

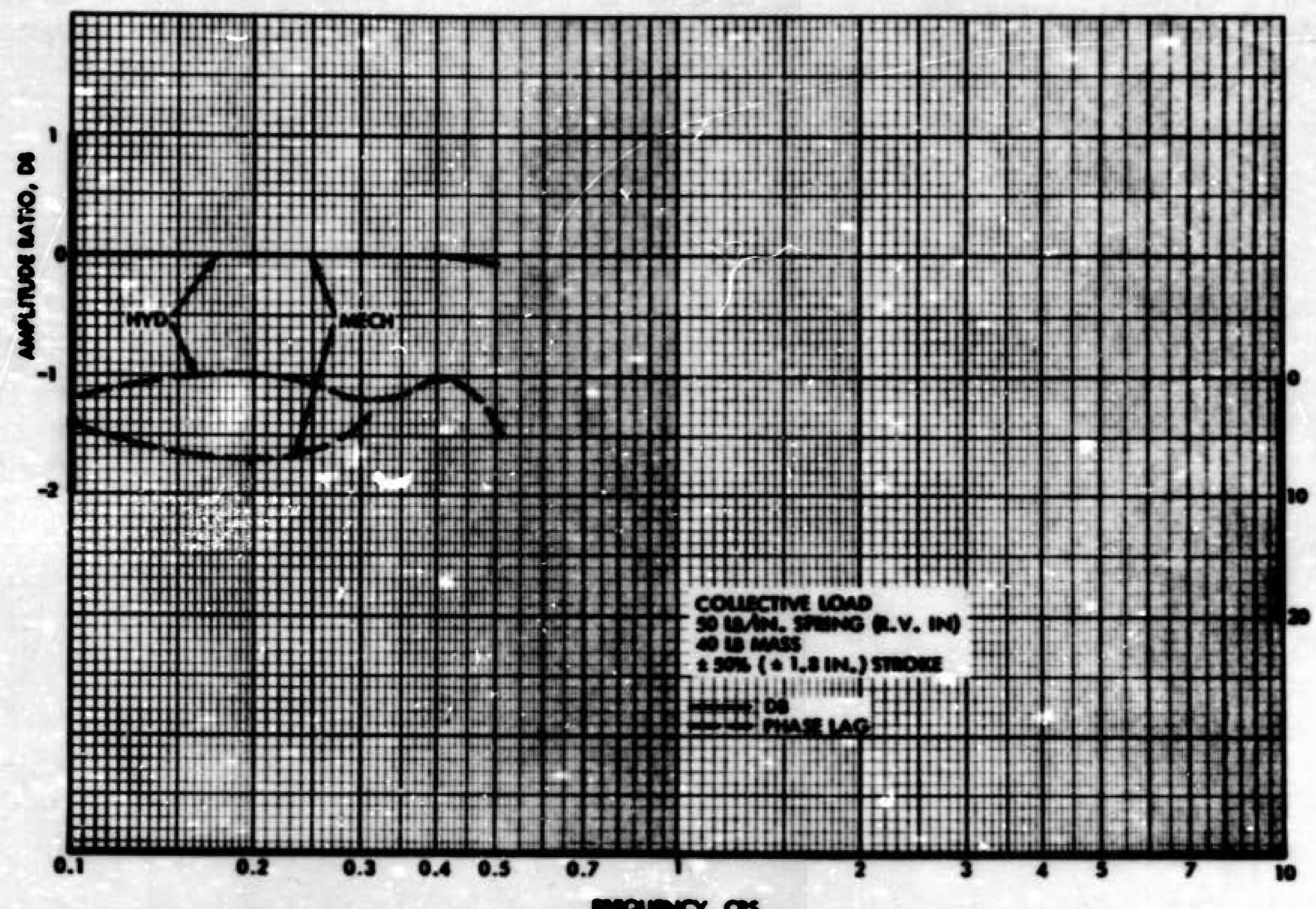

FIGURE 31 FREQUENCY RESPONSE, COLLECTIVE LOAD, 50-POUND SPRING, 40-POUND MASS, 50-PERCENT STROKE

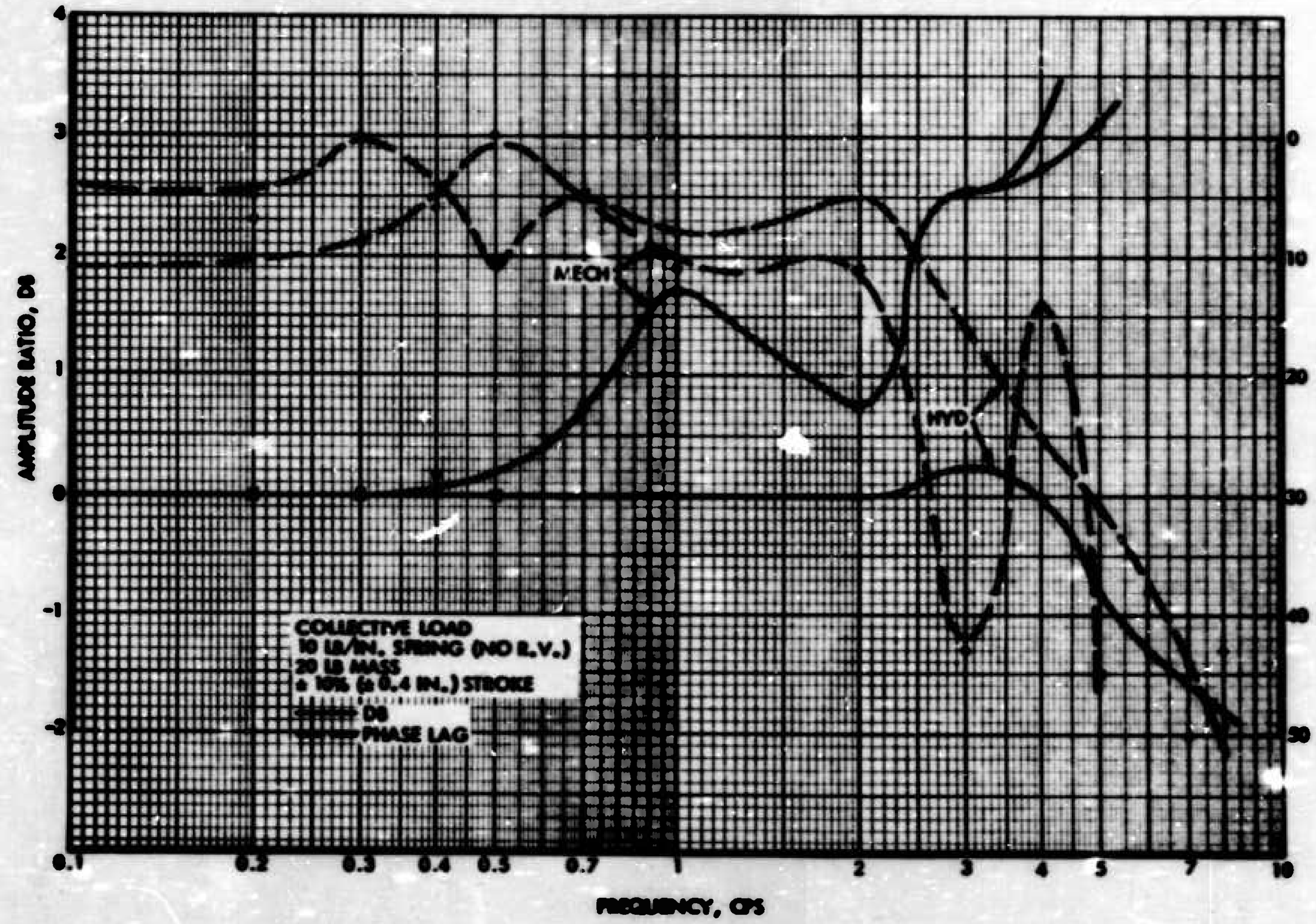

FIGURE 32 FREQUENCY RESPONSE, COLLECTIVE LOAD, 10-POUND SPRING, 20-POUND MASS, 10-PERCENT STROKE, (NO RELIEF VALVE) (AFTER ENDURANCE TEST)

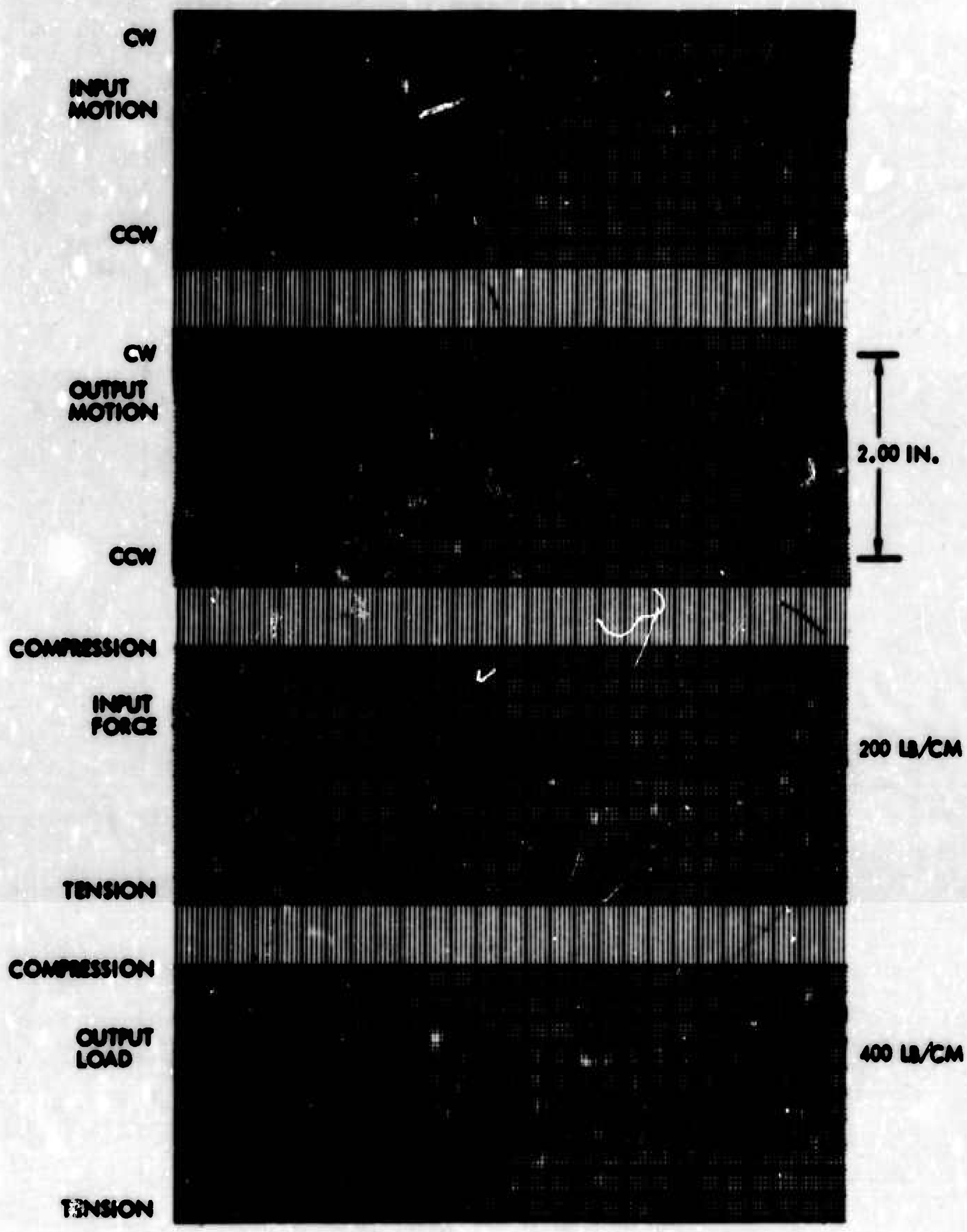

FIGURE 33 ENDURANCE CYCLING - MECHANICAL ACTUATOR - 510-POUND SPRING

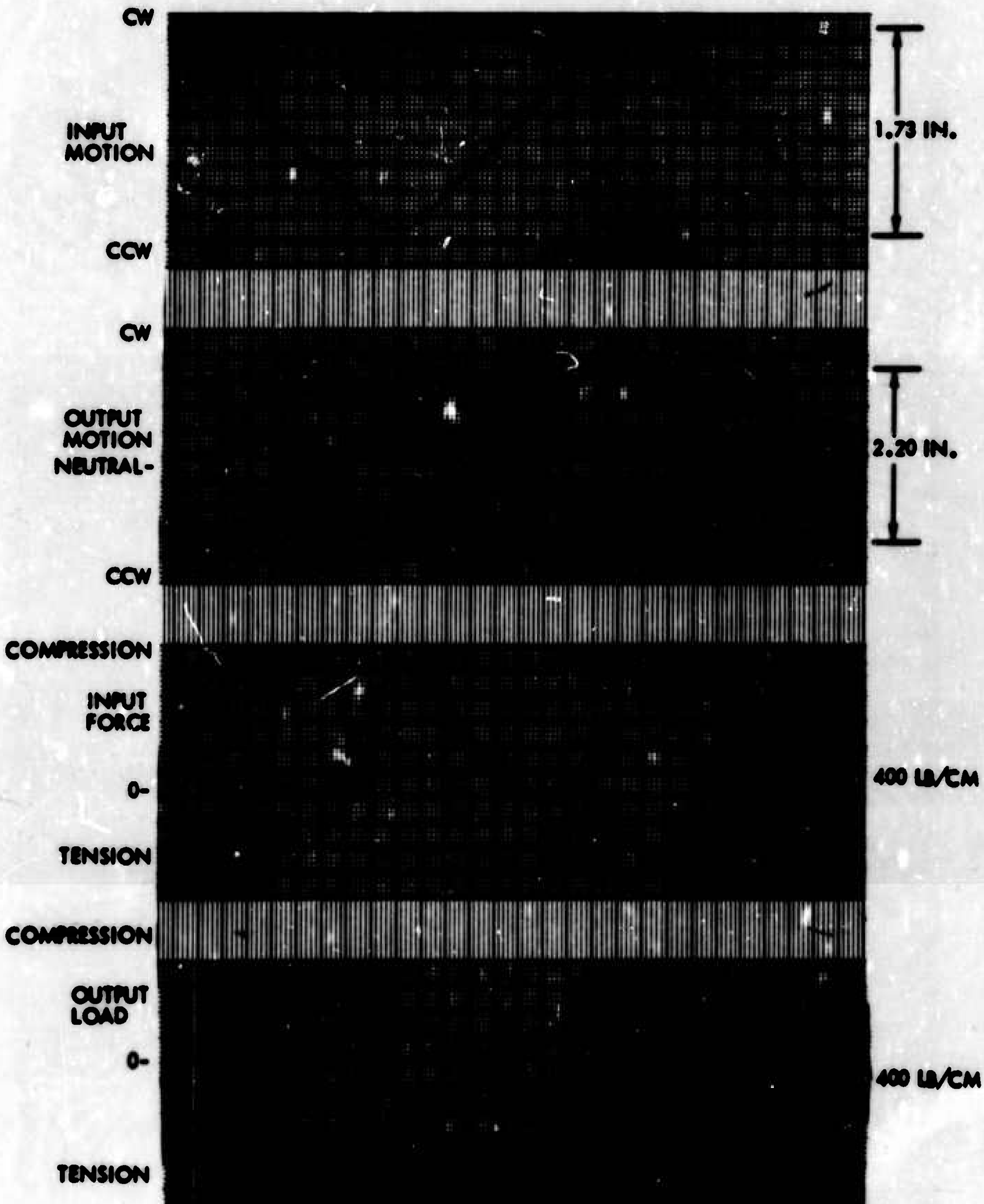

FIGURE 34 EMERGENCY OPERATION - MECHANICAL SERVO - CYCLIC LOAD - 510-POUND SPRING

output motion of the mechanical servo-actuator, as well as the input force and input motion of the input hydraulic cylinder. In addition, the test was repeated with collective loads of 10-pound/inch spring and 20-pound mass and the 50-pound/inch spring and 20-pound mass separately attached to the actuator while cycling through a full stroke of 4.00 inch. Figures 35 and 36 represent Sanborn traces of the actuator cycling.

HYDRAULIC SERVO-ACTUATOR

The hydraulic servo-actuator, Lockheed-California Company Part No. 540104-1, Serial No. 1102, was mounted in a loading test fixture as shown in Figures 37 through 40.

The two types of loads, cyclic and collective, were separately attached to the output yoke of the actuator housing. The various attached spring loads and masses were identical to those described in the mechanical servo-actuator section.

Following the initial failure of the bungee spring T-bracket of the mechanical servo-actuator, a complete series of cyclic load tests were performed on the hydraulic servo-actuator without relief valves installed in the plumbing lines to the input cyclinder. The results of the cyclic load tests are contained in the following identified paragraphs.

Cyclic Load - Force Threshold

The 510-pound/inch load spring was attached to the output yoke of the actuator,and the input to the control valve spool was disconnected. With a 1000-psi supply pressure applied to the hydraulic servo-actuator, the minimum input force required to obtain output motion of the actuator was measured. The test results were as follows:

Force Applied to Spool, lb	Direction of Actuator Motion	Type of Loading
0.22	Retraction	Tension
0.24	Retraction	Tension
0.24	Retraction	Tension
0.26	Extension	Compression
0.26	Extension	Compression
0.24	Extension	Compression

There was no noticeable change in the force required to move the actuator to any point along a ±1.00-inch stroke.

Cyclic Load - Resolution Test

The resolution of the hydraulic servo-actuator was determined by directly measuring the motion of the valve spool versus the initial output motion of the actuator body. Prior to this test, it was noted that the actuator

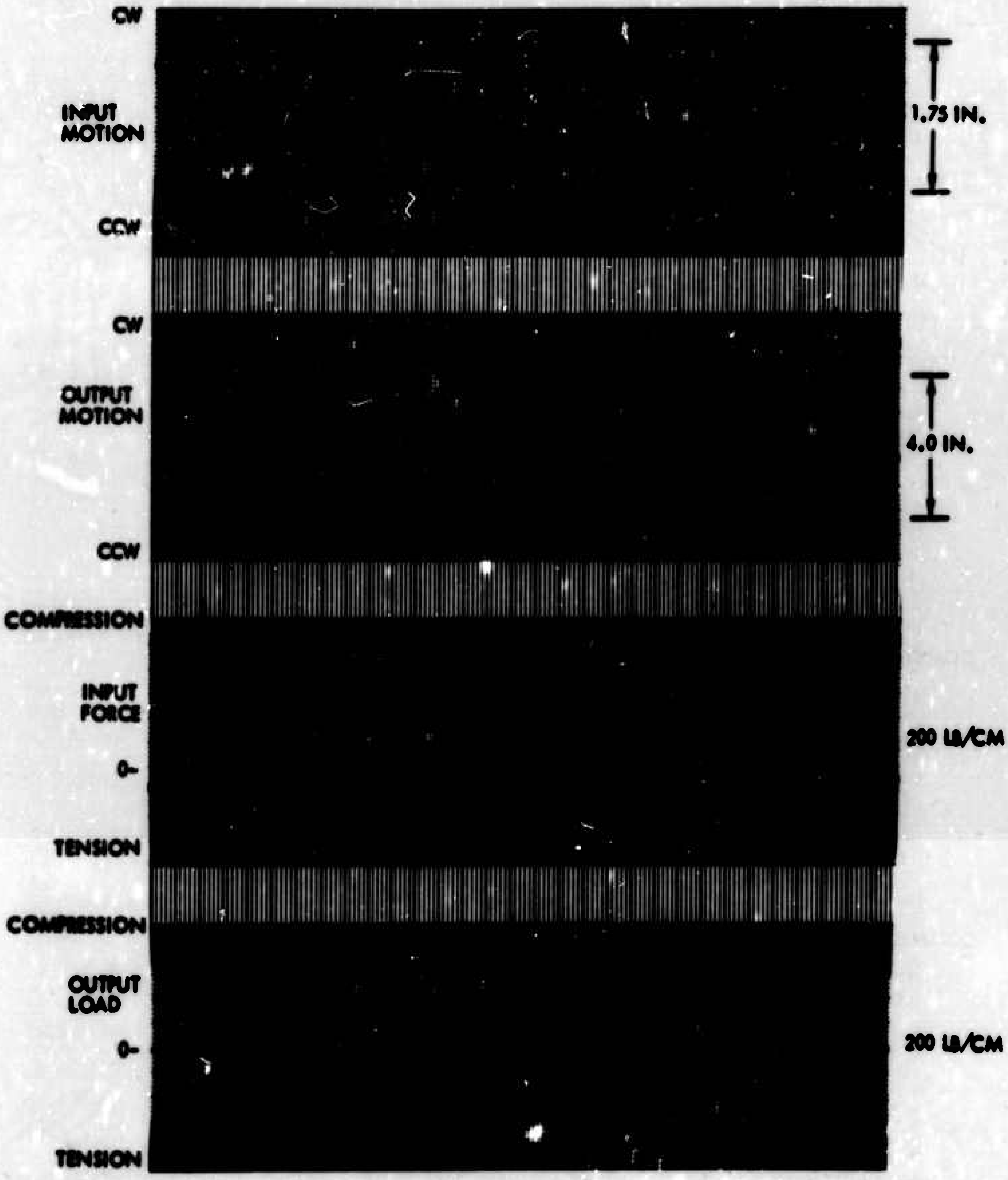

FIGURE 35 EMERGENCY OPERATION - MECHANICAL SERVO - COLLECTIVE LOAD - 10-POUND SPRING, 20-POUND MASS

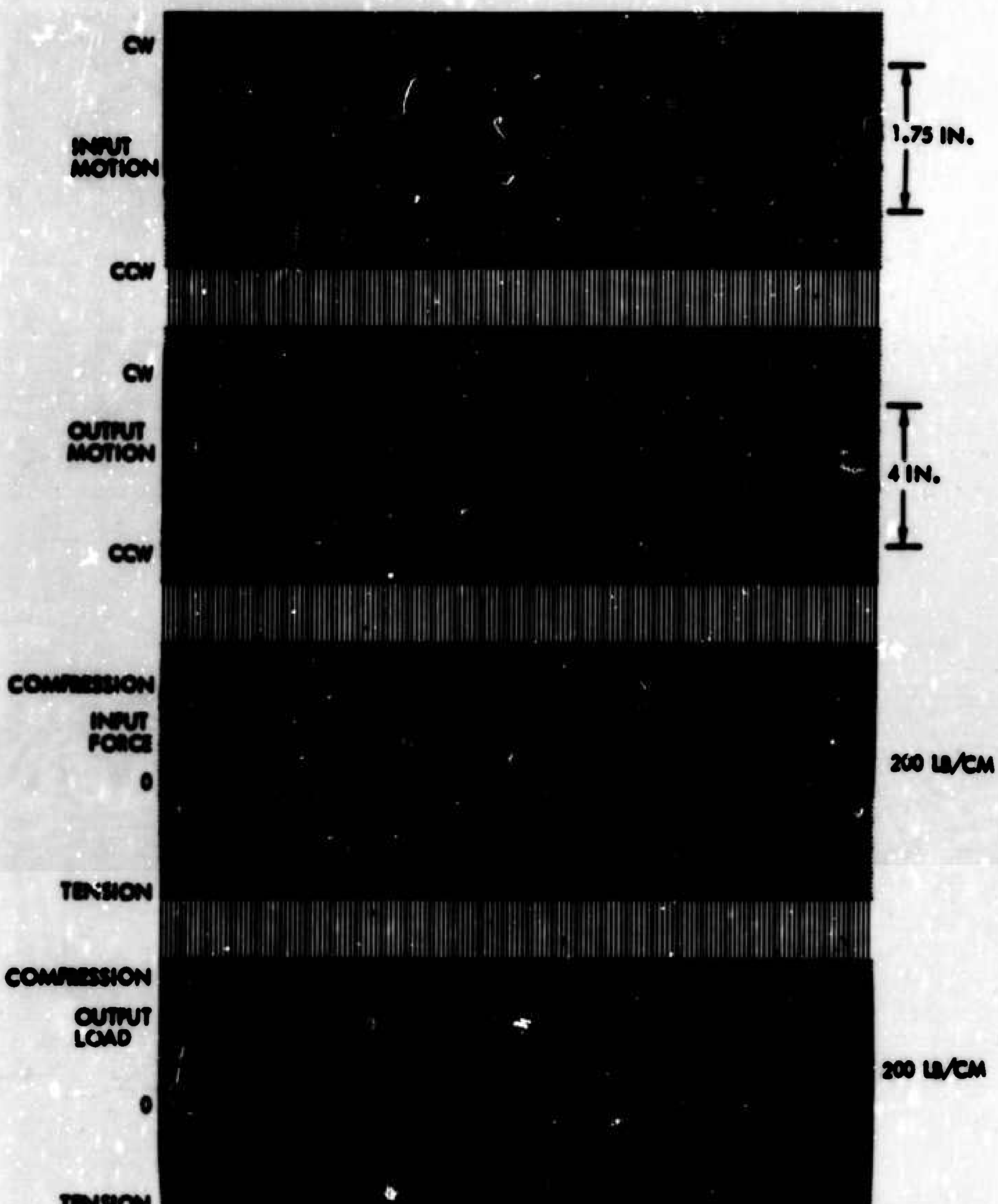

FIGURE 36 EMERGENCY OPERATION - MECHANICAL SERVO - COLLECTIVE LOAD - 50-POUND SPRING, 20-POUND MASS

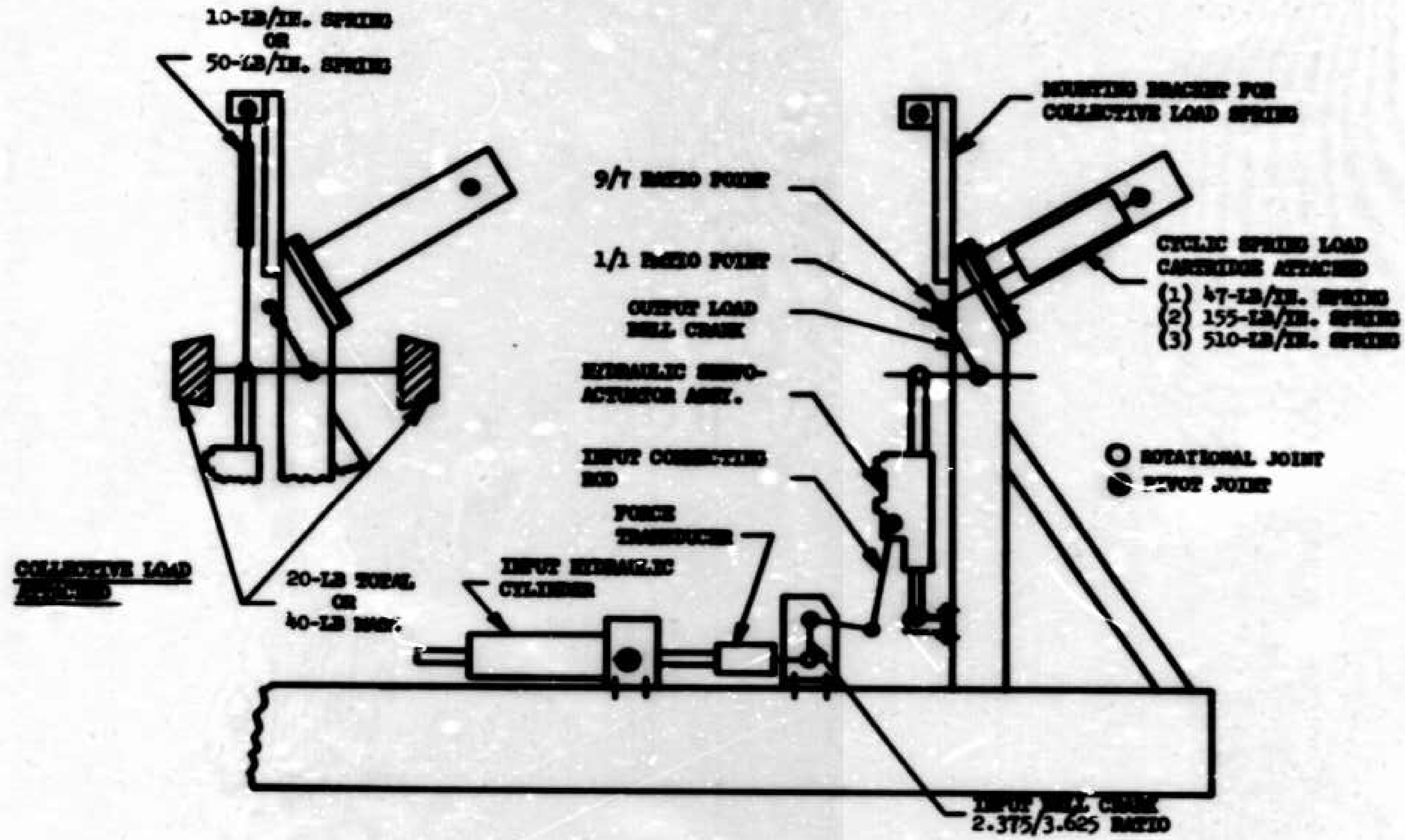

FIGURE 37 SCHEMATIC DIAGRAM - HYDRAULIC SERVO INSTALLED IN TEST FIXTURE

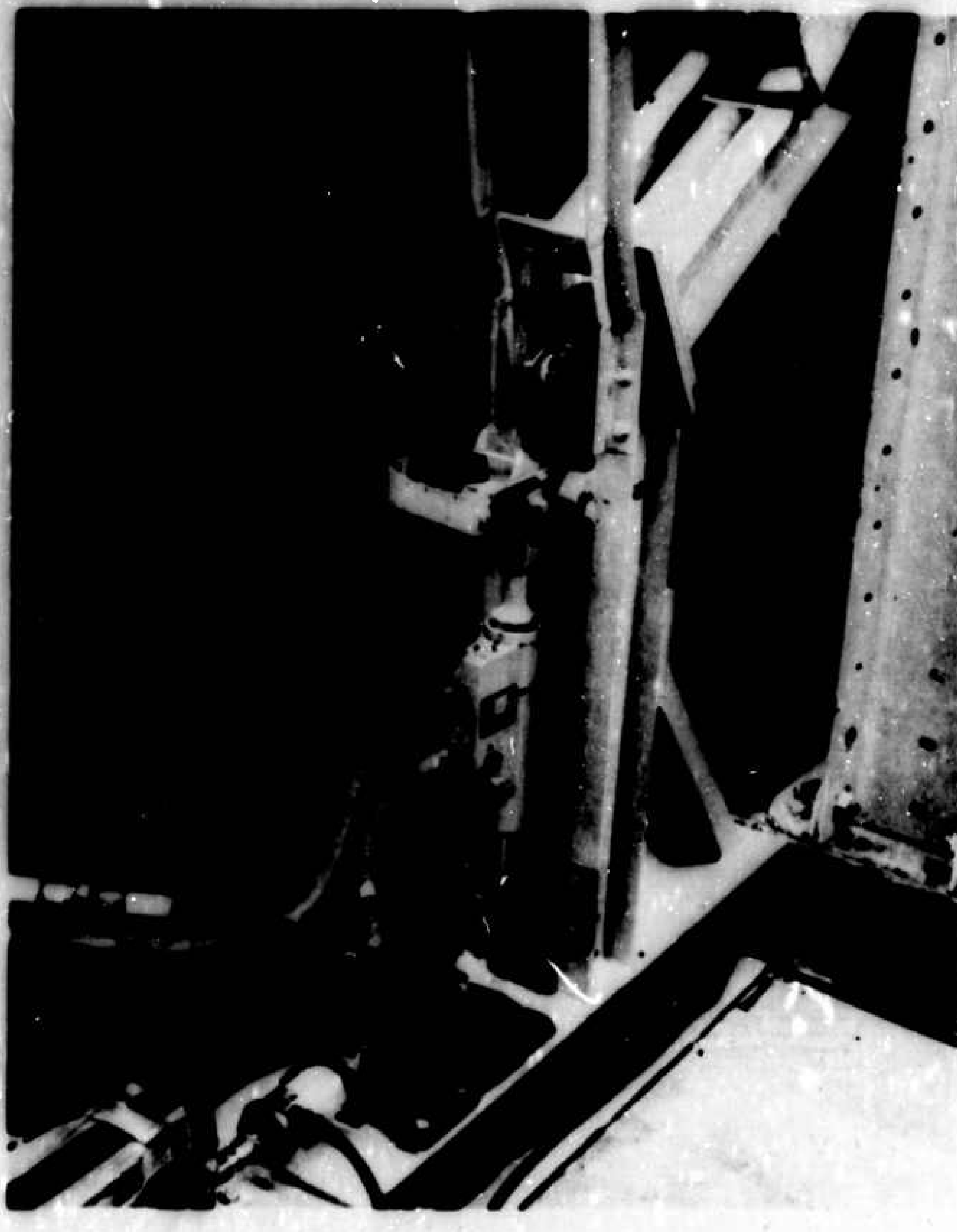

FIGURE 38 HYDRAULIC SERVO, CYCLIC LOAD, NEUTRAL POSITION

FIGURE 39 HYDRAULIC SERVO, COLLECTIVE LOAD, SIDE VIEW

FIGURE 40 HYDRAULIC SERVO, COLLECTIVE LOAD, GENERAL VIEW

had a dead-band stroke of 0.005 inch in the neutral (no load) condition only. When the actuator was applying an output load, there was no noticeable dead-band when using a 0.001-inch division dial gauge to detect motion.

An input command of approximately 0.0002 to 0.0003 inch was required to the valve spool to cause a noticeable output motion of the actuator body. Also, it was noted that the resolution did not change when any of the three cyclic load springs was attached.

Cyclic Load - Maximum Rate Test

The hydraulic servo-actuator was manually operated by moving the valve spool to its full open position in each direction with each of the three cyclic load spring separately attached. The results of these tests are tabulated in Table 3.

Cyclic Load - Frequency Response Test

The hydraulic servo-actuator was cycled sinusoidally at output amplitudes of ±0.05, ±0.10 and ±0.50 inch, at frequencies ranging from 0.1 cps to 10 cps. Frequency response measurements were recorded using each of the three cyclic load springs. After the completion of mechanical servo-actuator tests, the hydraulic relief valves were installed in the hydraulic lines of the input hydraulic cylinder and a frequency response test was performed with the 510-pound/inch load spring attached and at an output amplitude of ±0.10 inch. The results of those tests are presented in Figures 9 through 19.

Cyclic Load - Step Response Test

The hydraulic servo-actuator was operated at maximum rate with output steps of 0.10, 0.25 and 0.50 inch for each of the three cyclic load springs. The time constant was determined for each "step" condition in the same manner as described in the mechanical servo-actuator section. The results of these tests are tabulated in Table 4.

Collective Load - Maximum Rate Test

Prior to performing this series of tests, the mechanical servo-actuator had been completely tested. During this interval of time, the Serial No. 1102 hydraulic servo-actuator had to be returned to Lockheed stores; however, it was replaced with an actuator of the same type, Part No. 540104-1, Serial No. 103. Also, the hydraulic relief valves were installed in the hydraulic lines of the input hydraulic cylinder.

The hydraulic servo-actuator was operated by applying a square-wave electrical signal to the servo loop of the input hydraulic cylinder. This was performed for each of the four collective load conditions. Results of these tests are tabulated in Table 3.

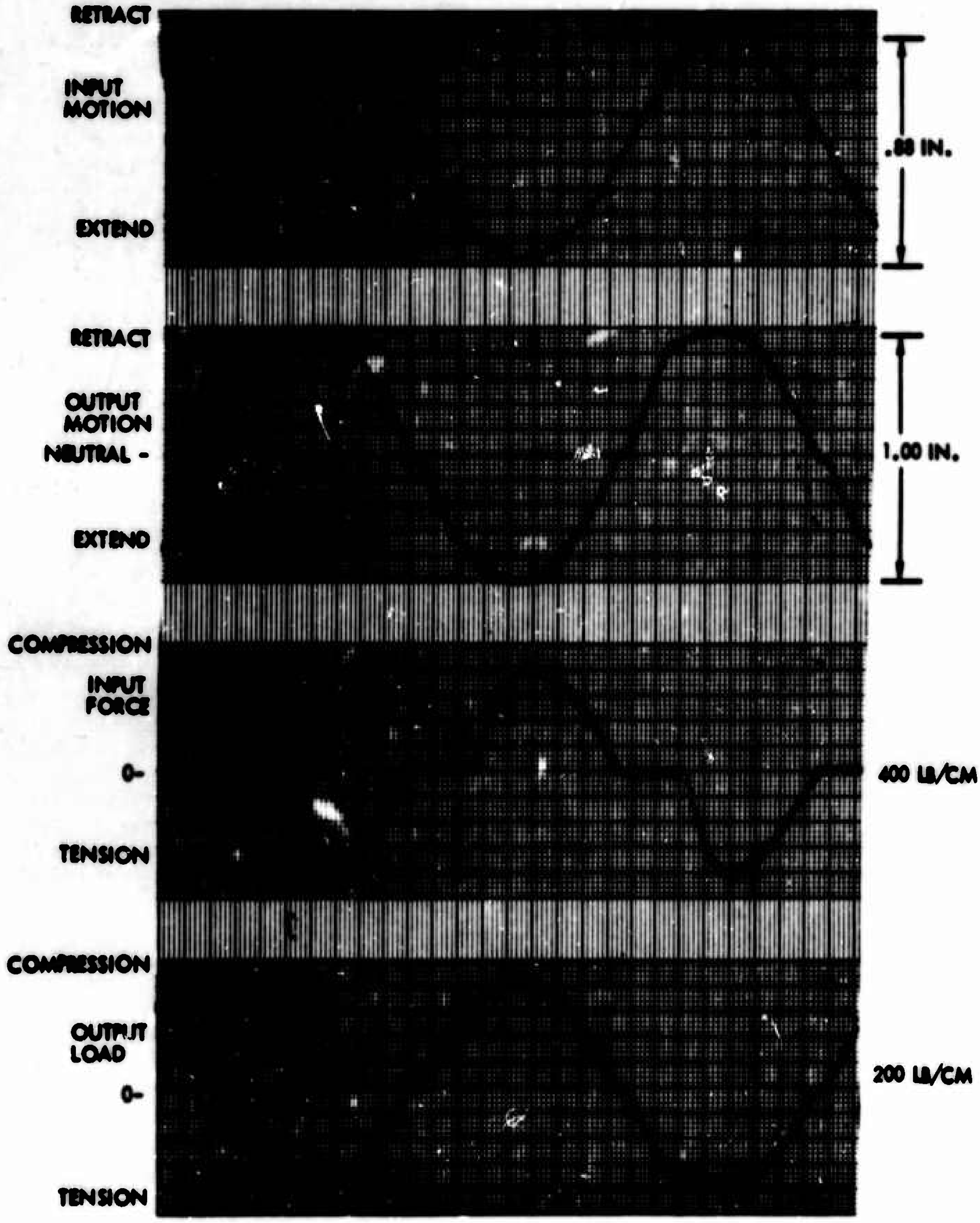

FIGURE 41 EMERGENCY OPERATION - HYDRAULIC SERVO - CYCLIC LOAD - 510-POUND SPRING

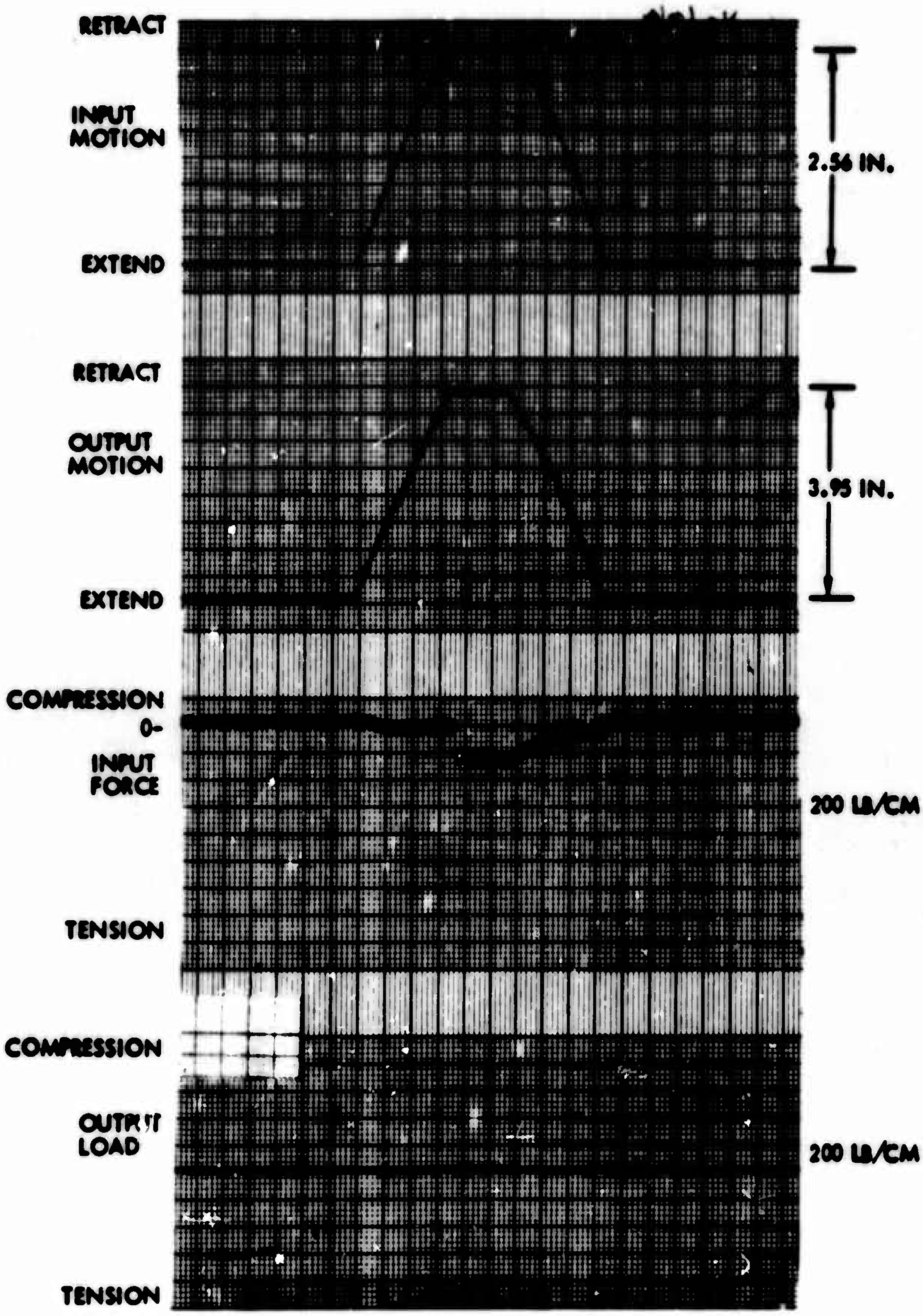

FIGURE 42 EMERGENCY OPERATION - HYDRAULIC SERVO - COLLECTIVE LOAD - 10-POUND SPRING, 20-POUND MASS

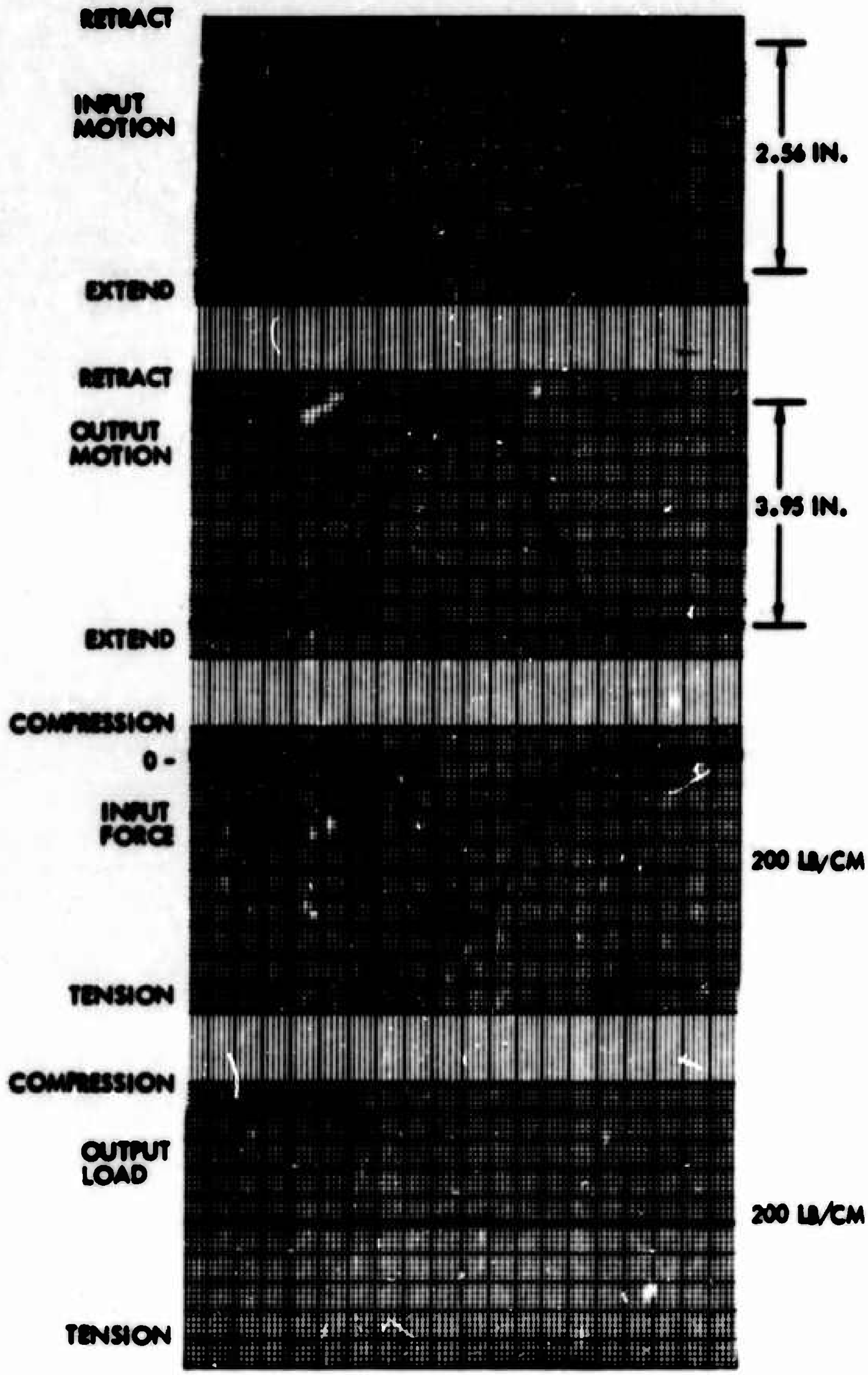

FIGURE 43 EMERGENCY OPERATION - HYDRAULIC SERVO - COLLECTIVE LOAD - 50-POUND SPRING, 20-POUND MASS

Collective Load - Frequency Response Test

The hydraulic servo-actuator was cycled sinusoidally in the same manner as described in the mechanical servo-actuator section. Results of these tests are shown in Figures 20 through 32.

Collective Load - Step Response Test

The hydraulic servo-actuator was tested in the same manner as described in the mechanical servo-actuator section. Results of these tests are tabulated in Table 4.

Emergency Operation Test

A loss of input hydraulic power to the hydraulic servo-actuator was simulated, thereby requiring a manual input force to operate the actuator with a 510-pound/inch cyclic load spring attached to the output yoke of the hydraulic servo-actuator. The actuator was operated through a stroke of 1.00 inch as commanded by the input hydraulic cylinder stroke of 0.88 inch. Figure 41 represents a Sanborn trace of the output load and output motion of the hydraulic servo-actuator as well as the input force and input motion of the input hydraulic cylinder. In addition, the tests were repeated with collective loads of 10-pound/inch spring and 20-pound mass and the 50-pound/inch spring and 20-pound mass separately attached to the hydraulic servo-actuator yoke. The actuator was cycled through a stroke of 3.95 inches simultaneously with the input hydraulic cylinder stroke of 2.56 inches. Figures 42 and 43 represent Sanborn traces of the actuator cycling.

RETEST OF MODIFIED ACTUATOR

Following the completion of the scheduled testing of the Curtiss-Wright mechanical servo-actuator, the test item was returned to the Curtiss-Wright Division of Curtiss-Wright at Caldwell, New Jersey, for investigation of possible changes to improve performance.

It was decided that the most expeditious change would be to reduce the gap between the inside diameter of the spring clutch and the outside of the rotating drum. This change would reduce the dead-band and also the force threshold; however, without additional changes, the braking capability of the spring clutches would be nullified.

The test actuator, modified as noted above, was returned to Lockheed, and some of the earlier tests were rerun to evaluate the effect of the modifications. The results are summarized as follows:

1. Dead-band. The input motion required to initiate output response from null position against the 510-pound/inch spring load was in the order of ±0.005 inch compared to the previously measured dead-band of ±0.050 inch. This represents a significant improvement.

2. Force threshold. A slight improvement in force threshold was noted at the null position (8 to 10 inch-pounds compared to the previously recorded 11 to 13 inch-pounds), but the force required to initiate further motion from a load-holding position was excessively high (30 to 40 inch-pounds). This was probably due to the fact that the load was not being held by a static brake but by energy from a slipping clutch spring. It can be concluded that the lack of a positive load brake on the modified unit is the reason for the high force level required to start motion against a high load. It should be noted that operation at a constant rate through full stroke against the 510-pound/inch spring load could be accomplished without a noticeable increase in input torque and that the reduced dead-band resulted in a smoother response, the amplitude of the "stair-step" output fluctuations being significantly reduced.

The results of this abbreviated retest indicate that redesign to provide a reduced dead-band will provide greatly improved performance and that design studies are required to minimize the input force levels

EVALUATION

The following paragraphs represent the contractor's evaluation of the test program performed under this contract.

INTERPRETATION OF TEST RESULTS

The testing of the mechanical servo-actuator under the type of loading provided by the test fixture, compared with the performance of a hydraulic servo under the same loading, indicated that overall performance of the mechanical servo was inferior to that of the hydraulic unit. In fairness, it should be pointed out that the hydraulic unit was a tried and proven production part with a basic design which had years of industry-wide development behind it; on the other hand, the mechanical servo is based on relatively new design principles, the test item was a prototype unit designed and built on a tight schedule and budget, and most of the moving parts were of new design. It is obvious that design improvements can be made to improve performance in the areas discussed below.

Dead-Band and Force Threshold

These two items are listed together, since the force threshold (the input force required to initiate output motion) is due primarily to the spring rate of the servo clutch and is therefore a function of the width of the dead-band. The dead-band in turn is determined by the clearance between the spring clutch and the constantly rotating power-driven drum.

The input torque, measured during the force threshold tests, ranged from 8 inch-pounds to 25 inch-pounds (see Table 1), with an average of approximately 11 inch-pounds at the null position. In comparison, the hydraulic servo required only 0.25-pound input force at the valve, which corresponds to an equivalent input torque of less than 0.5 inch-pound.

The maximum allowable breakout force permitted for cyclic control by MIL-H-8501A is 1.5 pounds at the stick. A design requirement for the actuator was a breakout force not greater than 50 percent of the MIL-H-8501 maximum. Converting 0.75 pound at 10 inches stick travel to a torque at 75 degrees angular travel gives an allowable force threshold of

$$0.75 \times 10 \frac{180}{75\pi} = 5.7 \text{ inch-pounds.}$$

The maximum allowable dead-band at the stick is ±0.2 inch. Half of this value is equivalent to 2 percent of total travel of 10 inches; 0.02 X 75 degrees = 1.5 degrees total allowable dead-band at the servo input.

The excessive width of the dead-band (a total of 2.5 degrees of input travel) has a pronounced effect on performance under a sinusoidal input. When the output load reverses direction, output position lags the input by as much as 1.7 percent of the total amplitude.

Modification of the original unit to improve these characteristics resulted in a significant reduction of the dead-band, but the force threshold was still higher than the allowable 5.7 inch-pounds.

Resolution

Tests performed to determine the smallest increment of input motion which would result in a corresponding output travel indicated that, against the 510-pound/inch spring load, the mechanical servo required input motion of at least 0.001 inch as compared to 0.002 inch for the hydraulic servo (see Table 2). Furthermore, the resolution was not consistent, especially when moving with the load. Redesign of the output brake element is necessary to correct this condition.

Noise Level

The high level of noise generated by the mechanical servo was especially noticeable under high-amplitude cycling against the heavy (510-pounds/inch) cyclic load spring. The basic cause of the noise is the release and re-engagement of the clutch spring each time the fixed-rate output motion catches up with an increment of input motion. The consequent fluctuation of the torque demand on the power input results in a corresponding fluctuation in the torsional strain of the flexible shaft which connects the electric motor to the servo. The resultant vibration of the flexible shaft and its housing was amplified by the plywood board to which it was attached for rigidity. Furthermore, the mounting of the test fixture to the laboratory floor provided negligible damping.

It is apparent that the noise level can be attenuated by careful attention to the design of the flexible drive shaft and its supporting structure and also to the mounting of the servo itself.

Endurance

The failure of the unit to complete the specified endurance tests without failure pointed out some of the weaknesses in the prototype design and therefore served its purpose. The failures were primarily in the attachment of gears to their shafts, and repairs to the prototype unit were easily made to allow completion of the endurance cycling.

Design changes are required to provide positive attachment of gears to shafts with provisions to preclude the possibility of axial slippage of keys.

Emergency (Manual) Operation

The first attempt to test the emergency (manual) mode of operation resulted in the structural failure of a small lug which transmits input forces to the input gearing through a C-shaped flat spring. The failed part was replaced, and the tests under normal powered operation were completed before repeating the manual operation test. The mechanical servo performed the emergency operation satisfactorily; however, the torque required to deflect the input bungee through half-travel seemed excessive. This torque was measured separately by a bench test of the input gear assembly and was recorded as approximately 30 inch-pounds at half-travel (37.5 degrees).

Output Rate

The results of the tests of maximum output rate indicated that the mechanical servo was capable of meeting the specified 5-inch/second rate requirements under the collective loading conditions in one direction only. However, the recorded rates were higher than those measured with the hydraulic actuator under the same loading.

With the cyclic loading, the mechanical servo rates were lower than the corresponding rates with the hydraulic actuator.

APPLICABILITY OF TEST RESULTS

This test program was intended to evaluate the performance of a mechanical servo-actuator for possible use in helicopter flight control systems, particularly a system similar to the XH-51A. The loads provided by the test fixture simulated the magnitude of the XH-51A cyclic and collective loads but did not accurately represent the actual dynamic conditions.

The test results indicate that the mechanical servo is best suited for functions which require output motion through a large amplitude at a constant rate. The frequency response data show that the mechanical servo, in its present state of development, cannot compete with a hydraulic servo in responding to small-amplitude commands applied sinusoidally at frequencies above 0.3 cycle/second. Furthermore, the resolution tests indicate that precise positioning of the output is difficult when motion is in the same direction as the load.

Inasmuch as the performance of the mechanical servo can be substantially improved by design changes in previously noted areas and since schedule and budget limitations did not permit a more accurate simulation of actual flight conditions, the results of this program should not be construed as a deterrent to further investigation of mechanical servo applications. It is suggested that follow-on programs be considered, using a whirl-stand to evaluate performance of a multi-axis system against a simulated dynamic rotor system.

There are significant potential savings in system weight by the use of mechanical servos in lieu of hydraulic servos in a vehicle of the XH-51A size. The existing hydraulic system weighs approximately 50 pounds, including the hydraulic actuators and a 10-pound emergency hydraulic system. Assuming that the gear box and flexible shaft required for power transmission weigh 5 pounds and that actuator weight can be limited to 8 pounds per unit, a system weight saving of 20 pounds can be realized. It should be noted that the prototype actuator weighs 18 pounds, but the designers have calculated that 8 pounds is a practical estimate for a repackaged unit.

The reliability of the mechanical servo and the power transmission elements must be demonstrated before its use in flight control systems can be considered. The limited endurance cycling performed during this program indicated that careful attention must be paid to the design of all small components. Furthermore, the tests indicated that dynamic load conditions cause peak power demands which can overload the power input elements. It appears likely that some form of torque-limiting clutch is required at the power input to preclude damage to the servo.

MODIFICATIONS DURING TESTING

During the test program, several modifications were made to the test setup and to the test unit. These changes, and the reasons therefor, were:

1. The inertia flywheel was eliminated from the power input to avoid time-consuming braking procedures required to stop flywheel rotation when it was necessary to shut off the power input.

2. The electric motor power drive was relocated after failure of a right-angle adapter for the flexible shaft drive. The revised setup utilized a straight adapter.

3. The mechanical servo was repaired at the start of the program to replace a small lug which failed during the initial manual operation check.

4. Relief valves were added to the hydraulic actuator which provided the input displacements to limit input load to 70 pounds, and the actuator was stroke-limited to avoid striking the mechanical servo input stops during full-amplitude cycling. Some tests were rerun without the relief valves to evaluate their effect on the results.

5. Repairs were made to the mechanical servo during the endurance test to permit completion of the specified test spectrum. These repairs are noted in Table 5.

PROPOSED REDESIGN

The results of the tests performed under this contract were reviewed by the Curtiss Division of the Curtiss-Wright Corporation, resulting in a submittal to Lockheed of Curtiss-Wright's ideas for a proposed redesign of the mechanical servo-actuator. The proposed redesign incorporates changes which provide a smaller, lighter, more serviceable package and features which will correct some of the noted performance deficiencies.

Size and Weight

Figure 44 shows a comparative size relationship between the test assembly 173410 and the proposed new design assembly. The use of a base casting, a reorientation of the spring clutches and bypass, and gearing design modifications allow for considerable assembly size reduction. The weight reduction is proportional to the envelope change and results in an estimated weight of 8 pounds for the new design as compared with 18 pounds for the test unit.

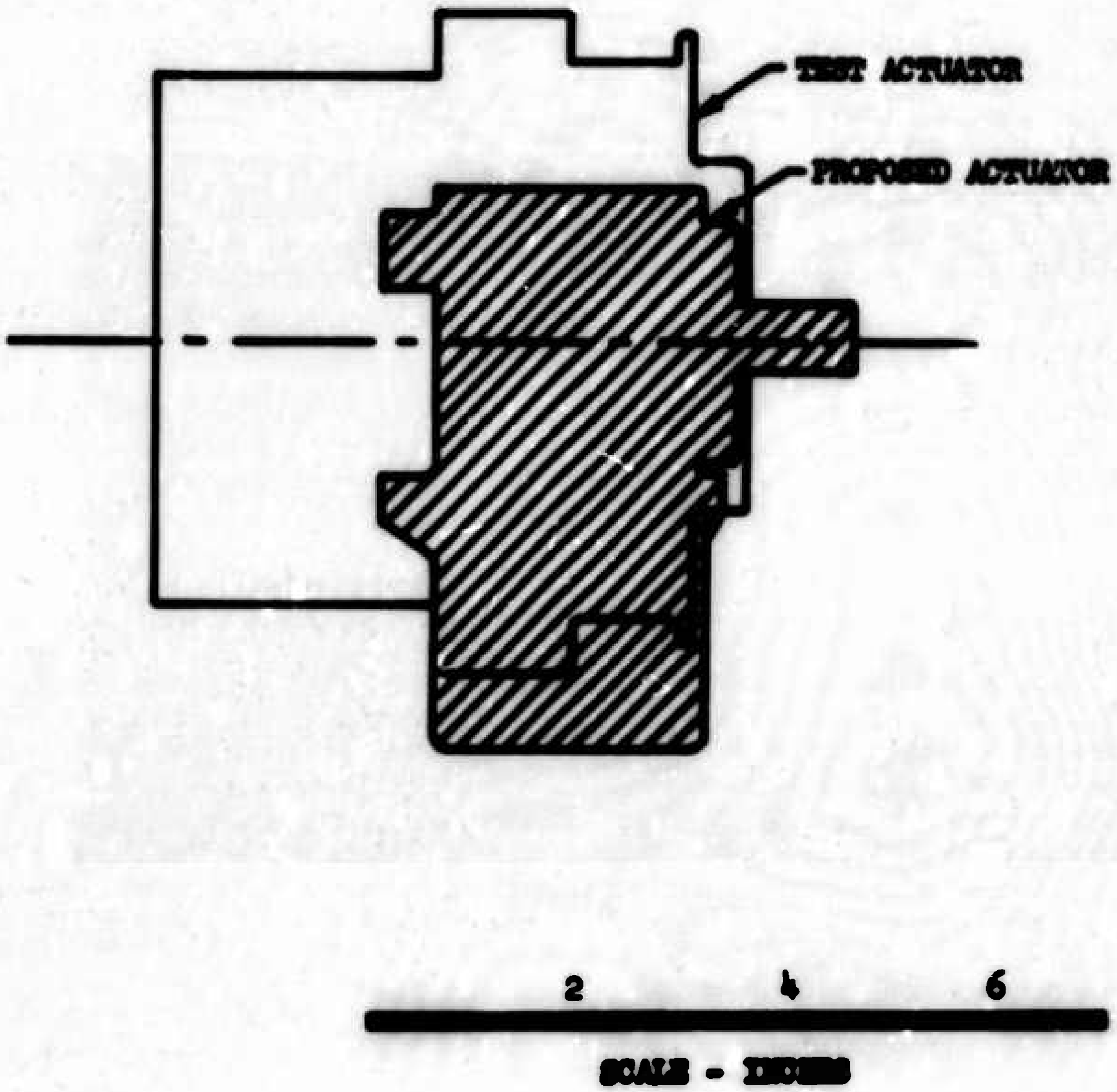

FIGURE 44 COMPARATIVE SIZE OF PROPOSED REDESIGN OF MECHANICAL SERVO-ACTUATOR

Serviceability

The new design utilizes a base casting to which are mounted the functional components. A cover appropriately sealed encloses the servo assembly. Removal of the cover exposes the operating components for inspection or service without major disassembly off the aircraft.

Functional Components

The 173410 test assembly incorporated a closed-loop follow-up servo system consisting of a spring clutch amplifier, a passive brake, a manual reversion bypass unit, power input gearing, signal step-up gearing, and output reduction gearing.

The proposed new design consists of a closed-loop follow-up servo system with spring clutch amplifier, an irreversible screw-jack brake, a manual reversion bypass unit, power input gearing, and simple signal and output linkages replacing the gear trains of the test unit.

A feature of the new design is that, in the new assembly, the input signal is transmitted through a bungee to a bar differential to which the output is attached. A mismatch of input and output position results in an error signal being transmitted through a transfer bearing to a helical gear shaft. The helical gears on this shaft are meshed with the clutch input and gears such that translation of the shaft results in relative rotation of the clutch input to output gears, causing the spring to expand to engage the power supply gearing. The clutch, thus engaged, drives the output through an irreversible screw jack and reduces the differential error to zero. The output power transmission is through the bypass similar to that in the 173410 unit. A mechanical connection of the input lever to the output lever through the bypass is similar to that in the 173410 unit, providing full mechanical control in any failure mode of the boost or power supply system.

Performance

The proposed revised design not only offers improvements in size, weight, complexity, and serviceability, but performance characteristic improvements are anticipated.

In the 173410 assembly, the springs used as clutches are also used as passive brakes. The major part of signal dead-band and breakout force is due to the necessity to unwind the spring from the braking surface, to wrap it through a space gap between brake and power drum, and to engage the driving drum. In the new design the springs are used only as clutches and are spaced closer to the drive drums such that angular rotation of the spring to engage the clutch is reduced to a minimum. The breakout force is reduced by virtue of the reduced torsional spring load during the engagement operation.

BIBLIOGRAPHY

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4. Test Procedure for Evaluation of Mechanical Servo-Actuator, Lockheed-California Company, Burbank, California, Report No. LR 18282, 19 October 1964.

APPENDIX

DESIGN SPECIFICATION FOR A MECHANICAL SERVO-ACTUATOR*

PREFACE

The design specification prepared as part of this contract is reproduced herein as it was submitted to the Curtiss Division of Curtiss-Wright Corporation, who designed and built a mechanical servo-actuator in accordance with the specification requirements.

Certain deviations from the specification requirements were discussed and mutually agreed to by Lockheed and Curtiss-Wright. These included (1) acceptance of the overall length of the actuator as 7.25 inches in lieu of 7 inches as specified by Lockheed drawing FCS-1000; (2) elimination of output travel stops; (3) agreement that breakout force and dead-band should be held to 50 percent of the maximum values specified by MIL-H-8501A.

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*Lockheed-California Company, Report No. LR 18027, 21 July 1964.

SCOPE

This specification outlines the design requirements for an item of equipment designated as a "mechanical servo-actuator", the purpose of which is to convert manual input displacements into proportional output displacements of specified power level, utilizing power from an available high-speed rotary mechanical source.

The actuator defined by this document is intended to perform any one of three independent functions relating to helicopter flight control. These are: (1) collective pitch (lift control), (2) longitudinal cyclic pitch control, and (3) lateral cyclic pitch control.

APPLICABLE DOCUMENTS

The following Government specifications and Lockheed-California Company drawings are applicable to the extent of their principle and intent as applicable and/or as specified herein. Wherever there is disagreement between the referenced documents and this specification, the requirements of this specification shall govern.

GOVERNMENT SPECIFICATIONS

MIL-F-9490B	Flight Control Systems; Design, Installation and Test of, General Requirements for
MIL-A-8064A	Actuators and Actuating Systems; Aircraft, Electro-Mechanical, General Requirements for
MIL-E-5272C	Environmental Testing; Aeronautical and Associated Equipment, General Specification for

LOCKHEED DRAWINGS

FCS-1000	Space Envelope and Structural Attachments - Mechanical Servo-Actuator

REQUIREMENTS

SPACE REQUIREMENTS

The actuator shall be designed to fit the space envelope and the structural attachments shown on Lockheed drawing FCS-1000, dated 17 July 1964.

TYPE

The servo-actuator shall be of the spring-clutch type. It will be tri-stable in character, have integral mechanical position feedback and respond to manual input signal demands.

OPERATIONAL ENVIRONMENT

The actuator shall be designed to operate at any temperature between -65°F and +250°F at any altitude from sea level to 50,000 feet.

SELECTION OF STANDARD PARTS

Wherever possible, AN or MS standard parts shall be used.

Commercial parts having suitable properties may be used in applications where no suitable AN or MS standard parts are available.

MATERIALS AND PROCESSES

Material

The component parts of the actuator shall be fabricated from material which meets the quality requirements of applicable government specifications.

Protective Treatment

Where materials are used in the construction of the servo-actuator that are subject to deterioration when exposed to climatic and environmental conditions likely to occur during service usage, they shall be protected against such deterioration in a manner that will in no way prevent compliance with the performance requirements of this specification.

DESIGN REQUIREMENTS

Power Input

The servo-actuator shall be designed to accept a rotary mechanical power input with a nominal speed of 6300 rpm and a rated torque of 10 inch-pounds at that speed. The power input elements should be designed to with-withstand a limit stall torque of 18 inch-pounds.

Signal Input

The servo-actuator shall be designed to accept signal inputs applied in either direction by a pushrod attachment to a rotary crank. The angular excursion of the input crank shall be limited by stops to a total travel of 75 degrees, ±10 degrees. The input crank shall be designed to accept input displacements of 1.88 inches total for the cyclic function and 2.80 inches total for the collective function.

Output Load Characteristics

The servo-actuator shall be designed to control output position in response to input signals against two basic types of loading:

1. The output load for the cyclic function consists of a spring force proportional to output displacement working against gyroscopic inertia and viscous damping. The spring force is zero at mid-stroke and has a gradient of 665 pounds/inch in either direction over a displacement of ±1.02 inches. The load transmitted through the spring acts against a damping force of 25 pounds/inch/second and against a gyroscopic inertia load having an angular momentum of 222 slug feet^2/second at an effective moment arm of 2.83 inches.

2. The output load for the collective function consists of mass inertia, aerodynamic load proportional to displacement, and negligible damping. The mass inertia is 20 pounds at a moment arm which provides an output travel of 4.35 inches. The aerodynamic load can vary from a zero gradient to a gradient of 50 pounds/inch.

Output Travel Limits

The output crank travel shall be limited by adjustable stops to a range compatible with the input travels and the load characteristics specified above under Signal Input and Output Load Characteristics, respectively.

Emergency Operation

The servo-actuator shall be designed to permit actuation of the output by manual force applied to the signal input crank in the event of failure of the normal power source. Dead-band and hysteresis under emergency operation shall be held to a practical minimum.

Structural Requirements

The servo-actuator shall be structurally capable of withstanding a limit load of 1600 pounds applied to the input crank at the cyclic input attachment and a limit load of 1000 pounds applied to the collective input attachment, in any crank position, in a direction normal to the crank in mid-position. The actuator shall be irreversible under a static loading of 1000 pounds in either direction at the output attachment for the cyclic function and a static loading of 600 pounds at the output attachment for the collective function.

Service Life

The servo-actuator shall be designed for an operational service life of 1000 hours without overhaul. Provisions shall be made for necessary relubrication as required.

PERFORMANCE REQUIREMENTS

1. The servo-actuator shall be capable of providing an average output rate of 5 inches per second at the collective function attachment

point for the full range of travel against the inertia loading specified in the preceding paragraph entitled Output Load Characteristics, paragraph 2.

2. The servo-actuator shall be capable of providing an average output rate of 2 inches per second at the cyclic function attachment point for travel from mid-position to either extreme against the spring load, inertia and damping specified in the preceding paragraph entitled Output Load Characteristics, paragraph 1.

3. Under the loading conditions specified in paragraphs 1 and 2 above, the servo-actuator shall be capable of providing output position control in increments of 0.01-inch maximum for the cyclic function and 0.02-inch maximum for the collective function.

4. The anticipated load-cycle spectrum for the cyclic function (based on 1000 hours) is as follows:

Amplitude (%)	No. of Cycles	Frequency (cps)
100	1,000	0.10
50	5,000	0.50
10	100,000	5.00

INTERCHANGEABILITY OF COMPONENTS

Components of the servo-actuator which may require replacement during the test program shall be designed to be physically and functionally interchangeable.

DIMENSIONS

The overall size of the servo-actuator shall not exceed the space limitations of Lockheed drawing FCS-1000. It is desirable to have the input and output attachments in a common place, but separation of input and output planes will be permissible if necessary to avoid compromising other requirements of this specification. It is desirable to have the power input located as shown on Lockheed drawing FC-1000, but alternate locations may be acceptable.

WEIGHT

Maximum weight of the servo-actuator shall not exceed 8 pounds.

FINISH

Protective finish in general accordance with specifications MIL-S-5002-2 and MIL-F-7179A shall be provided for all components of the servo-actuator. Necessary lubrication of moving parts shall be provided. The color of external finish on exposed parts is optional.

IDENTIFICATION OF PRODUCT

The servo-actuator shall be marked for identification with a nameplate in general accordance with the requirements of MIL-STD-130.

WORKMANSHIP

The workmanship of components and assembly shall be in accordance with standard practice for airborne equipment of this type.

ENVIRONMENTAL REQUIREMENTS

The servo-actuator shall be capable of meeting the following test requirements defined by specification MIL-E-5272:

Acceleration:	Procedure I
Vibration:	Procedure XII
Shock:	Procedure IV
Sand and Dust:	Procedure I
Humidity:	Procedure I
Salt Spray:	Procedure I

QUALITY ASSURANCE PROVISIONS

ACCEPTANCE TESTS AT SOURCE

Prior to delivery of the servo-actuator to the Lockheed-California Company, sufficient tests shall be performed to verify that the general requirements stipulated in the paragraphs entitled Design Requirements and Performance Requirements, previously discussed, have been satisfied. The Lockheed-California Company shall be advised where and when the testing will be performed and invited to witness the tests.

ACCEPTANCE TESTS AT LOCKHEED

Upon receipt of the servo-actuator, Lockheed-California Company will perform acceptance tests based on the requirements referenced immediately above. The specific acceptance test procedures will be included as part of the test procedure scheduled for completion seven weeks prior to the scheduled delivery date of the servo-actuator (see following Contractual Test Program).

REJECTION AND RETEST

If the servo-actuator fails to meet the specified requirements during acceptance tests, an investigation of the extent of and the reasons for the discrepancy will be made jointly by the manufacturer and the Lockheed-California Company. Corrective action may be proposed by the Lockheed-California Company with due consideration for the effects on cost and schedule.

CONTRACTUAL TEST PROGRAM

A five-week test program, to be conducted at the Lockheed Rye Canyon Research Center, is scheduled to start within one week after receipt of the servo-actuator. The test procedure is scheduled to be completed (by the Lockheed-California Company) seven weeks prior to the delivery date of the servo-actuator. The manufacturer shall provide technical assistance to Lockheed during the testing phase

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U.S. ARMY AVIATION MATERIEL LABORATORIES

13. ABSTRACT

A prototype mechanical servo-actuator was tested under simulated loads to determine its suitability for use in powering flight control systems for helicopters. Tests included frequency response, step response, threshold, hysteresis, endurance, and emergency operation, and were compared with similar tests of a typical hydraulic servo. Recommendations are made regarding design changes for further investigation.

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UNCLASSIFIED

Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
ACTUATORS, MECHANICAL						
TESTING - CONTROL SYSTEMS						

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